ACOUSTIC POWER DEVICES

2nd Edition

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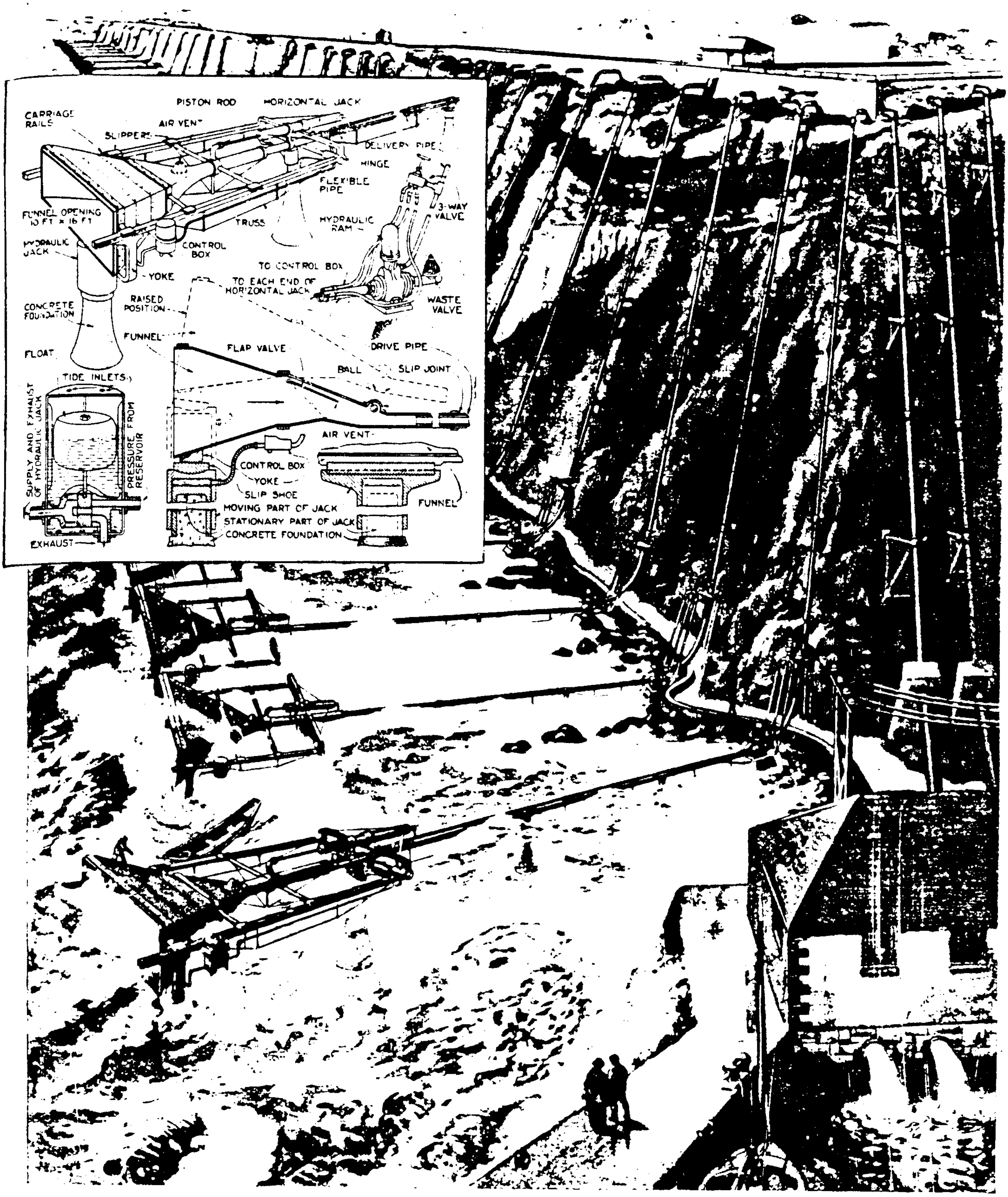

Sea Waves to Drive Novel Power Plant

TO LET the ocean waves do the work of man by providing hydroelectric power, Lieutenant Commander Lybrand Smith, of the Navy Department's Bureau of Engineering, has designed an amazingly ingenious battery of hydraulic rams, illustrated by this schematic drawing and the accompanying diagram.

The drive of the waves into the great funnels forces the water through pipes up the cliff into the long flume along the edge of the cliff. This flume carries the water into the reservoir, from which the accumulated water pours down larger pipes with sufficient force to operate the generating machinery in the power house below.

The hydraulic rams operate hydraulic jacks which are simply valves so arranged that the water forced into them by each successive wave does not fall back when the wave's strength is spent, but is locked in to remain until more water is forced in by the next wave. Each valve, like a door that opens but one way, is pushed open by the water's force. When the force is exhausted the weight of the water closes the valve, holding it shut until the next wave presses hard enough to open it.

Toribio Bellocq: Wave Pump

The Bellocq Wave Pump was the first new mechanical principle to be discovered in the 20th century (Wilkes' Rolamite was the second). It defies Toricelli's Law by utilizing pressure waves rather than suction to pull water higher than 30 feet (the pump is located at the surface, not at the bottom of the well). The technology is very efficient and requires little power. In practice, it is possible to expel during each stroke of the piston a volume of liquid about 17 times as large as the volume displaced by the piston (about 5,000 liters/hour in the preferrred embodiment described in the patent).

Plus: US Patents for very similar acoustic wave pumps invented later by Arthur P. Bentley & Albert G. Bodine.

Thanks to RexResearch.com

(*Popular Science*, ca. 1930)

"New Pump Beats Natural Laws in Raising Water"

Following the example of the United States Patent Office, 18 countries have issued patents to an Argentinian inventor upon an amazing pump that seems to violate natural laws. By creating waves in a pipeful of water, it makes the liquid run uphill.

When the inventor, Toribio Bellocq, applied for a US patent on a pump to be mounted on top of a well and to draw water up from almost unlimited depths, officials pointed out that his device apparently would have to defy the law of gravitation. Every high school student knows that by no effort can a pump suck water higher than approximately 33 feet. This is the limit at which the weight of an imprisoned column of water balances the atmospheric pressure outside. To force water higher from its source, authorities have always agreed that it must be pushed from below. Therefore Bellocq's "wave pump" seemed in a class with perpetual motion machines, which are not patentable because they are impossible.

Bellocq built one of his pumps, installed it atop a Washington DC office building, and invited officials to inspect it. They saw it draw a steady stream of water up a pipe 80 feet high. Not until they dropped weights down the pipe and found no unseen machinery did they believe their own eyes. Then they acknowledged that Bellocq had chanced upon an entirely new mechanical principle and issued his patent.

So extraordinary is the operation of the new wave pump that even Bellocq admits that he is not certain of its principle , and leaves to scientists the verification of his own explanation.

In Bellocq's pump a piston vibrates rapidly with an extremely short stroke. It deals hammerlike blows to a column of water in a pipe. His theory is that when the frequency of the blows is properly timed for the length of the pipe, a series of "stationary waves" is set up.

Suppose the pipe's bottom to be closed, then layers are formed where the water is alternately rarefied and compressed without moving. Midway between these and at the bottom are regions where water rushes alternately up and down because of the waves.

When a one-way ball valve is added at the bottom, water enters from outside at one point in each wave cycle, to replace water moving upward from the bottom of the pipe. Once inside, it cannot back out. Every influx of water "inches" the whole column upward, without interfering with the waves that travel through it. A valve at the outlet, while not essential, improves the efficiency.

The Acoustic Power Patents
# of Albert Bodine Jr., 1944-1988

DISCONTINUED--all US Patents are available online at no charg

Volume One

1. R. H. Dickinson, "Pump for Liquids," 1

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Pneumatic Options
Grass Valley, California

The Acoustic Power Patents of Albert G. Bodine Jr., 1944-1988

Volume Two

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Pneumatic Options
Eugene, Oregon

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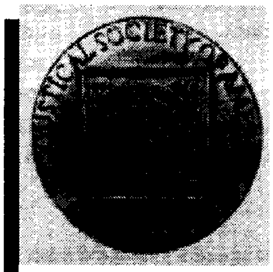


The Silver Medal

The Silver Medal in Engineering Acoustics

The Silver Medal is presented to individuals, without age limitation, for contributions to the advancement of science, engineering, or human welfare through the application of acoustic principles or through research accomplishments in acoustics. The recipients of the Silver Medal in Engineering Acoustics are listed below.

Year	Recipient	Contribution
1974	Harry F. Olsen	For his innovative and lasting contributions in microphones, loudspeakers, sound reproduction, and electronic music, his many publications, and his constructive editing.
1976	Hugh S. Knowles	For leadership, innovation, vision in the application of acoustical science and technology in industry and government and, in particular, for contributions to the advancement of technology for hearing improvement.
1978	Benjamin B. Bauer	For his contributions to engineering acoustics, particularly in the development of techniques and devices used to pick up, record, and reproduce sound.
1982	Per Vilhelm Bruel	For significant contributions in sound level instrumentation and precision measurement, and for notable leadership in international standards in acoustics.
1984	Vincent Salmon	For contributions in the design of horns and the control of noise and vibration.
1986	Albert G. Bodine	For his ingenuity in developing sonic vibratory devices of great technological importance.
1989	Joshua E. Greenspon	For his leadership and contributions to the solution of underwater radiation and scattering problems.
1992	Alan Powell	For leadership in research in the silencing of ship noise and for fundamental contributions to aeroacoustics.
1995	James E. West	For developing and optimizing polymer electret transducers.
1998	Richard H. Lyon	For contributions to noise reduction and products through design and to Statistical Energy Analysis.
2001	Ilene J. Busch-Vishniac	For development of novel electret microphones and of micro-electro-mechanical sensors and positioners.

Arthur P. Bentley: Sonic Pressure Wave Pumps

5

Calgary Herald (Tues., Oct. 10, 1989)

"Special Pump Pushes More Than Profits"

by

Richard Bruner

(*Christian Science Monitor*)

Tucson, AZ ~ Juan Pasco is a retired United nations development specialist who not only has found a solution to water problems of the world's poor rural areas, but has helped put together a Tucson corporation to make a lot of money in the process.

Pascoe's black eyes blaze with excitement when he talks about a new pump that will push water out of the ground at the rate of two gallons a minute, using a one-fifth horsepower that derives its energy from four solar panels.

"It's a godsend", he says.

Pascoe is the president of a Tucson company that will manufacture and market the new pump.

The pump is the invention of a senior genius who lives in Crossroads. Arthur Perry Bentley is the grandson of the man who created the car bearing his name.

Bentley's pump uses a motor that would barely drive a sewing machine. But its work is very limited. All it needs to do is send sonic waves down the well pipe to a valved pumping unit at the well's bottom. The unit picks up the waves and drives its piston up and down. This gushes a whole column of water up the pipe.

None of this takes much energy, Pascoe points out. Only enough to make some noise down a pipe.

Bentley has also invented a second type of pump, one that slides an electric charge down a black well pipe. The entire pipe becomes like a capacitor of a discharge system of an electronic ignition. It employs its valve system to push the column of water or oil to the surface.

Pascoe discovered Bentley in the mountains near Ruidoso, NM, where he was living a quiet existence, despite the 34 patents in his name. With Allyn Spence, an anthropologist with the Office of Arid Land Studies at the University of Arizona, Pascoe put together a company called **Appropriate Technology Development Inc**.

No long after forming the company, Pascoe and Spence took a prototype pump to the Navajo reservation.

"We put it in an abandoned well, connected it to four solar panels, and it started to bring up water", says Pascoe.

"The Navajo came around. The women especially were enchanted. They said, 'The sun is bringing out the water from the ground? Impossible."

For Pascoe, the experience on the Navajo reservation was confirmation of his hope that the pump might have worldwide uses.

6 "Knowing what I know about the needs of the developing world rural people without water or with very little water, I know one of the reasons they cannot use traditional pumping gear is because they don't have electricity or it is too expensive. Now here is a pump that the deeper you go the more efficient it is. You can go 4,000 feet and bring water up with a little motor that draws practically no electricity."

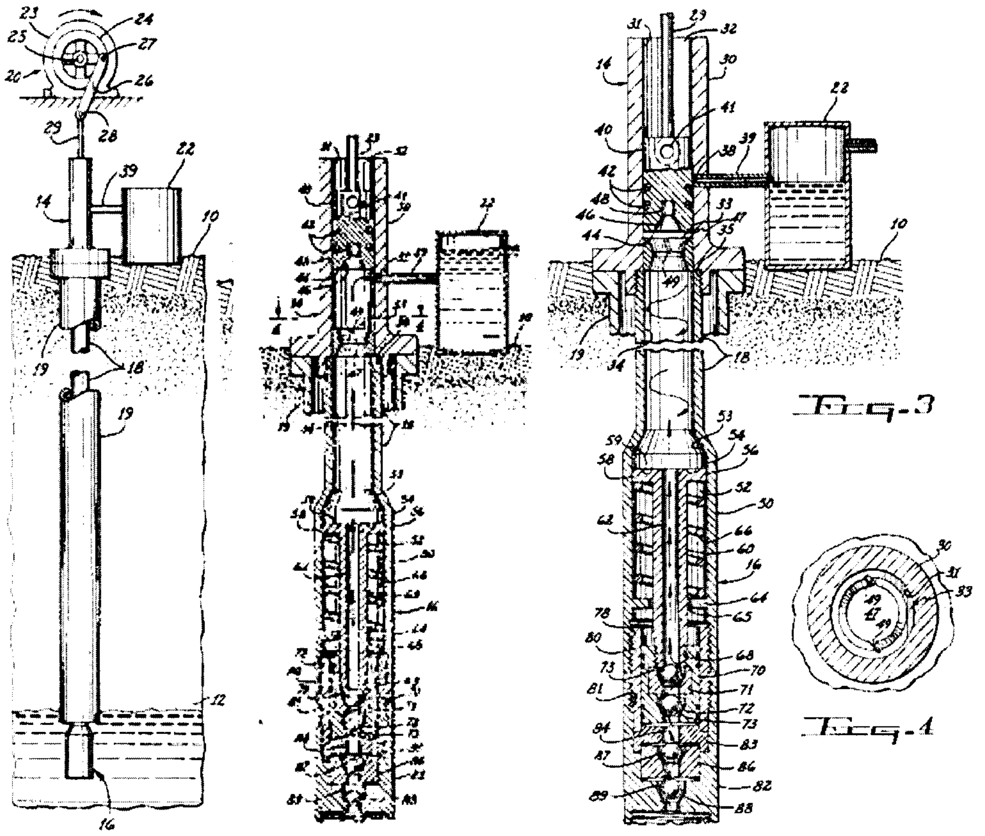
The Bentley pumps are also much less expensive to purchase and maintain than other water-pumping technologies. The smaller pumps will replace $20,000 windmills on the Navajo reservation at an initial per pump cost of $7,000.

The impact of such savings could be profound. The USDA has concluded that the biggest expense for farmers who experienced serious drought in the last two years is the cost of pumping water. That cost has driven more farmers into bankruptcy than any other single factor.

Appropriate Technology Development will work with China to have a joint-venture manufacturing operation in Shanghai. And it also will start making pumps in Tucson sometime soon.

US Patent # 4,295,799 (Oct. 20, 1981)
Sonic Pressure Wave Surface-Operated Pump
Arthur P. Bentley

Figure 1 Figure 2 Figure 3 Figure 4



Abstract ~

A single tube surface operated pump including a piston reciprocally mounted in a cylinder for alternately opening and closing a lateral fluid delivery port and for generating a sonic

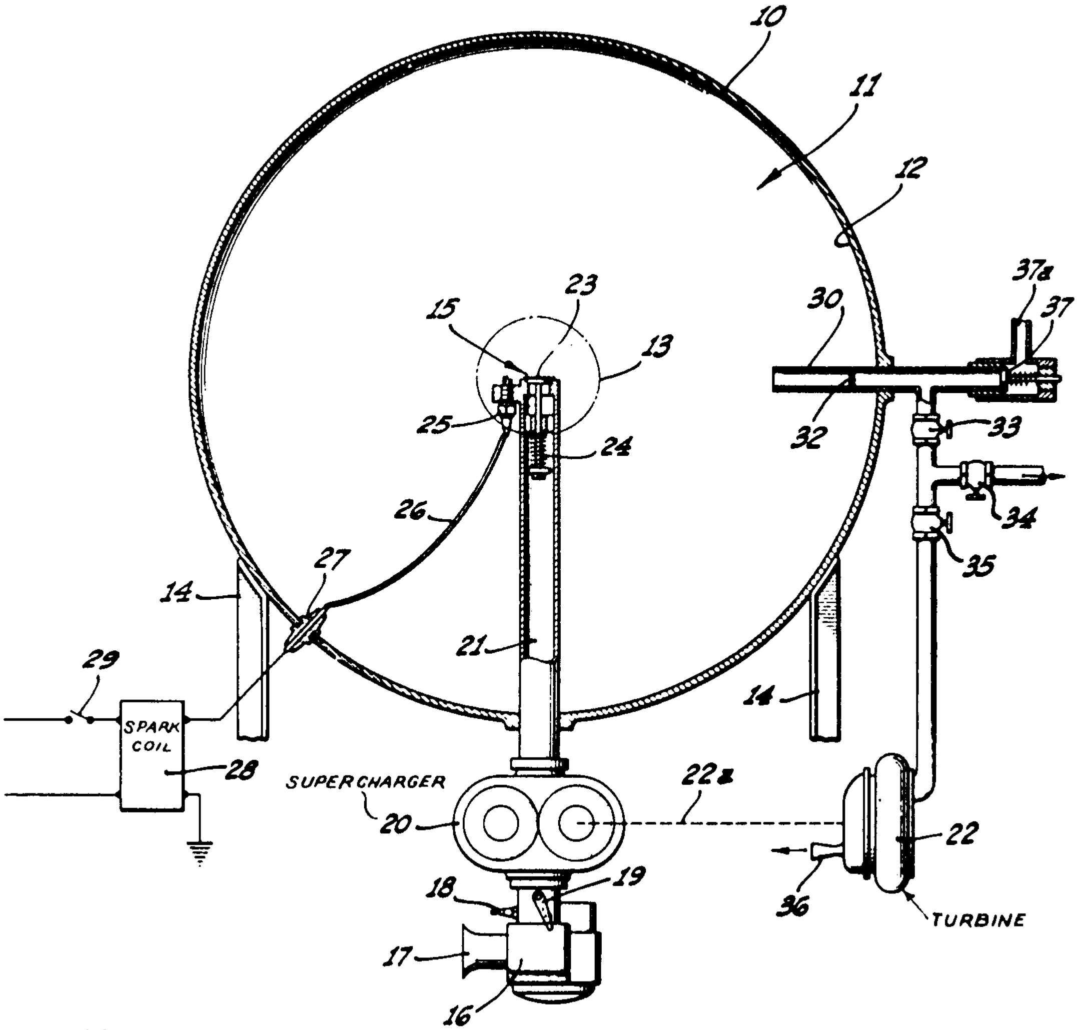
pressure wave by impacting a column of fluid in a metallic tube extending from the cylinder to a remote pumping mechanism located in communication with the fluid to be pumped. The piston is especially configured with a central recess in the face thereof so that the sonic pressure waves generated thereby will pass through a sonic nozzle and move downwardly toward the pumping mechanism in a spiral-like motion against the inner wall of the metallic tube and enter into a sonic intensifier chamber and are reflected off the pumping mechanism into a central column which travels back toward the cylinder and causes the fluid to be pumped to move in that same direction.

Jan. 8, 1952 A. G. BODINE, JR 2,581,902

RESONANT GAS COMPRESSOR AND METHOD

Filed April 23, 1945 6 Sheets-Sheet 1

FIG. 1.



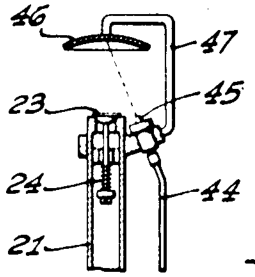


FIG. 2.

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RESONANT GAS COMPRESSOR AND METHOD

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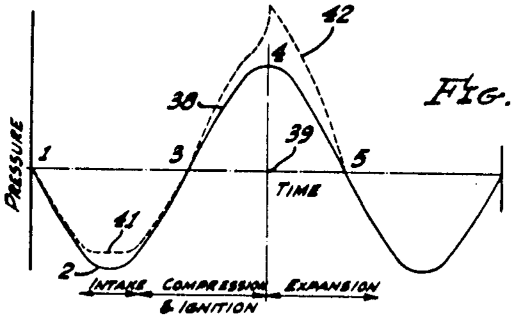

FIG. 4.

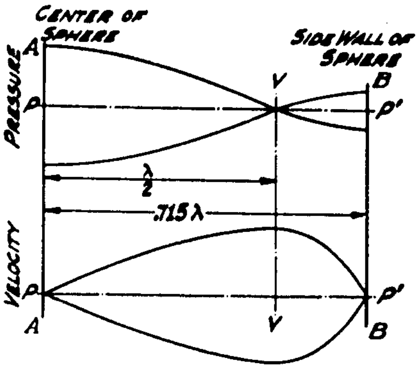

FIG. 3.

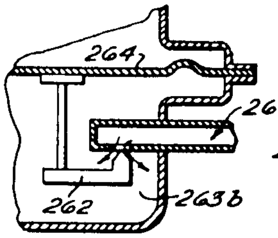

FIG. 3a.

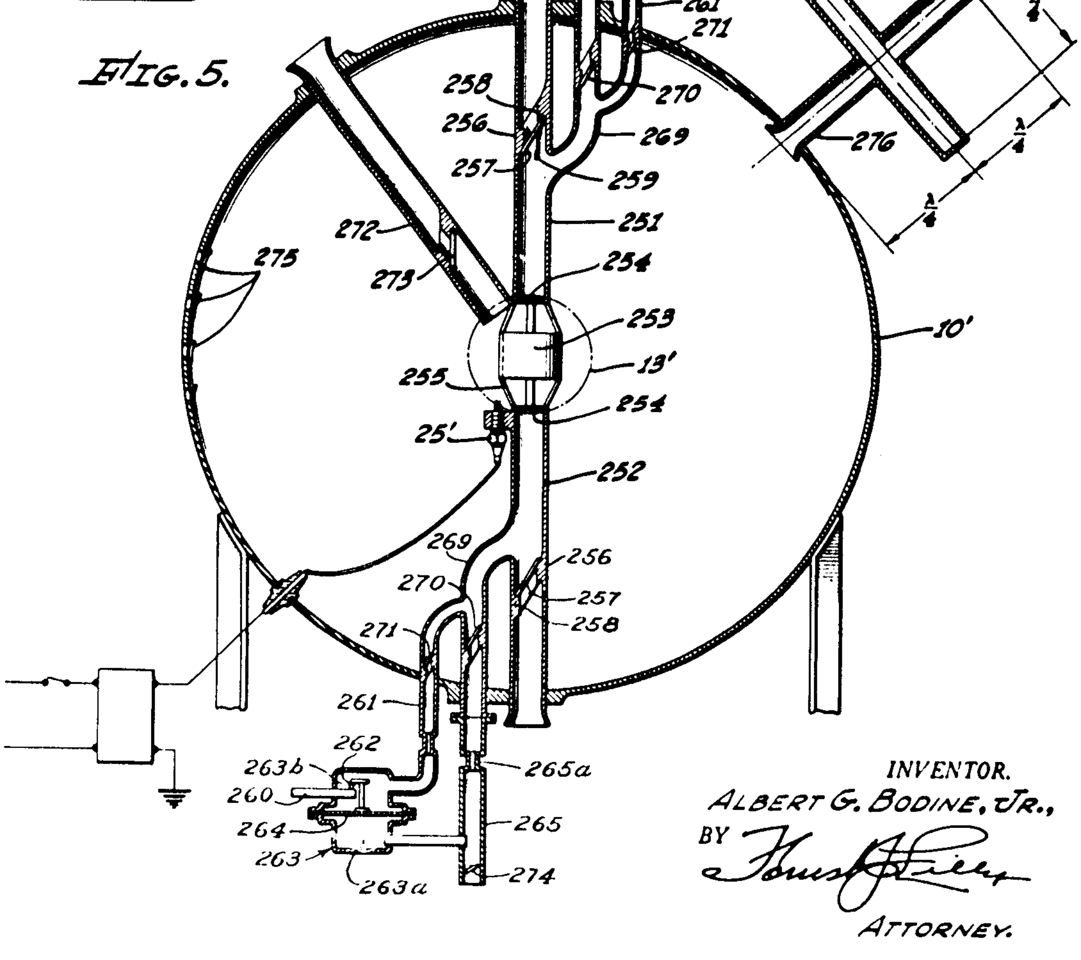

FIG. 5.

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RESONANT GAS COMPRESSOR AND METHOD

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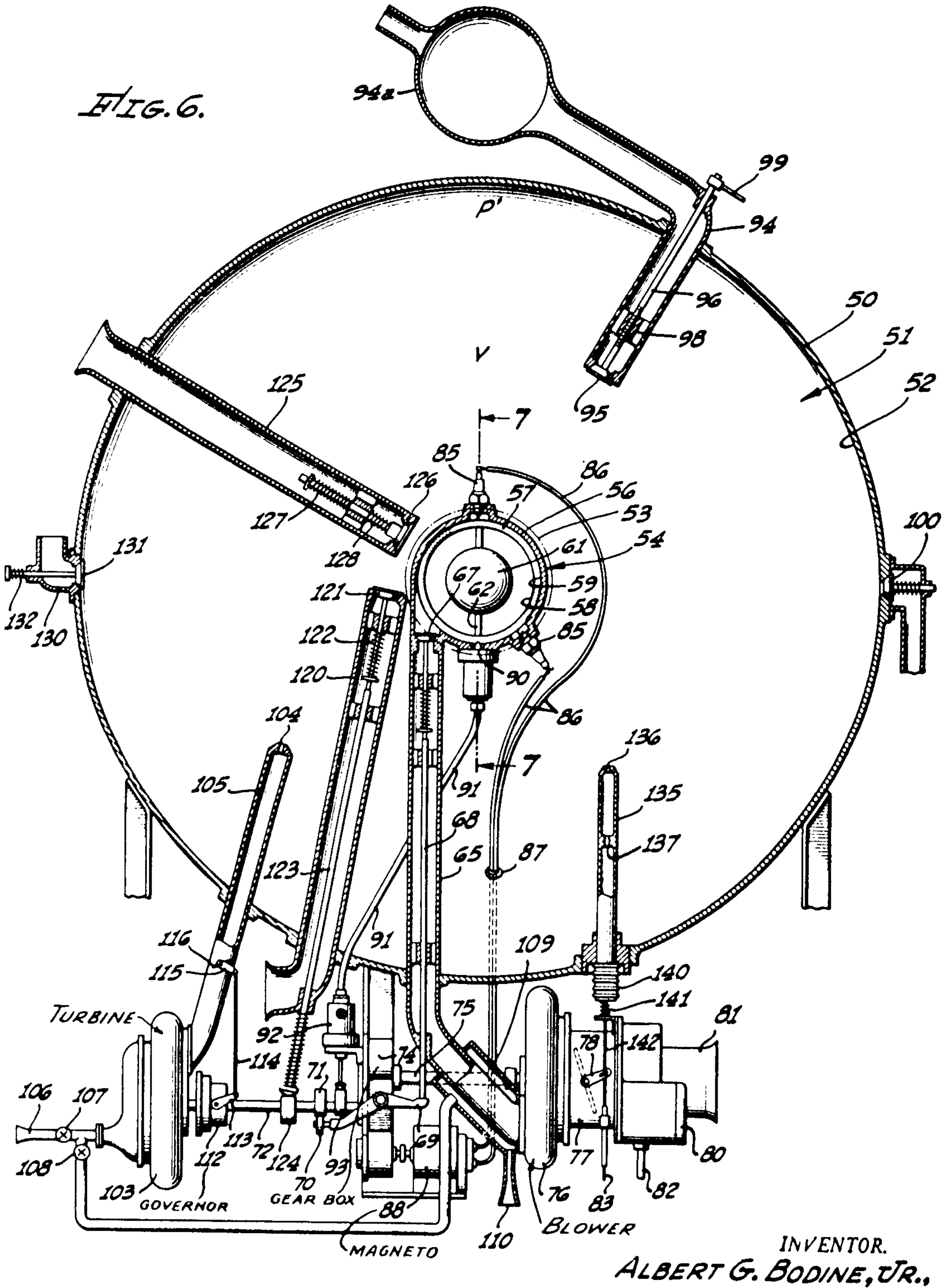
FIG. 6.



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RESONANT GAS COMPRESSOR AND METHOD

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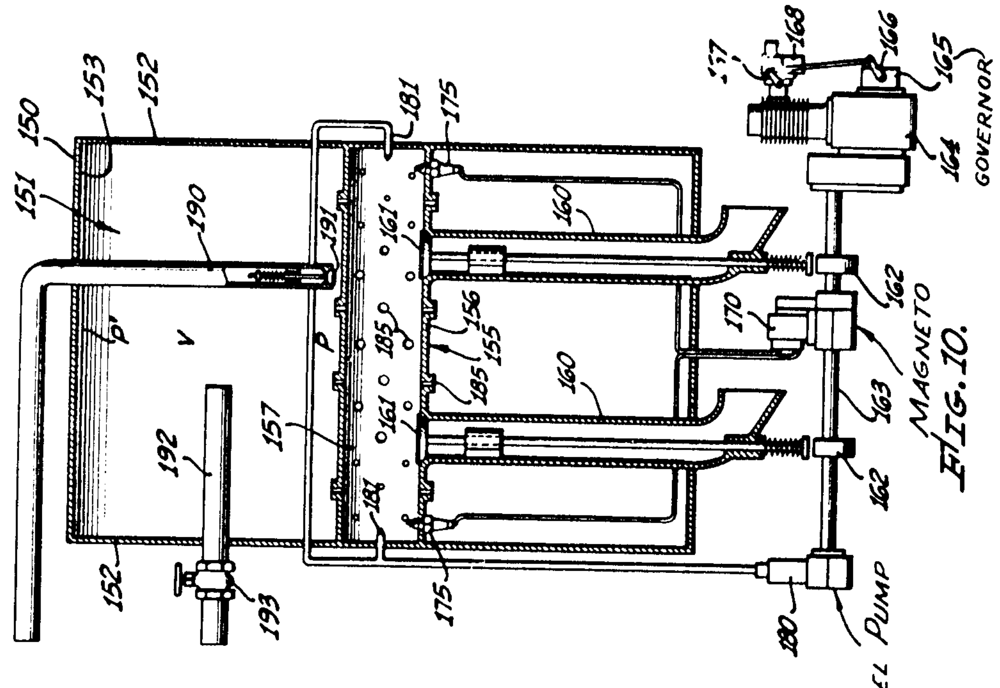

FIG. 10.

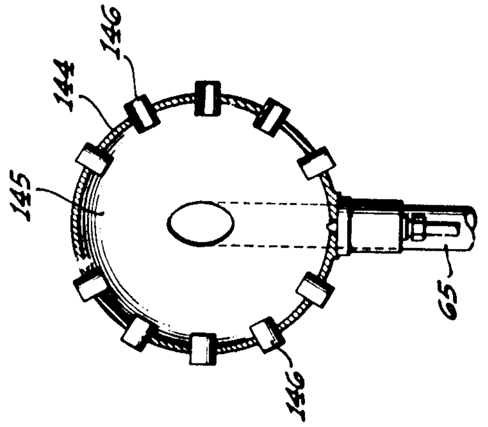

FIG. 9.

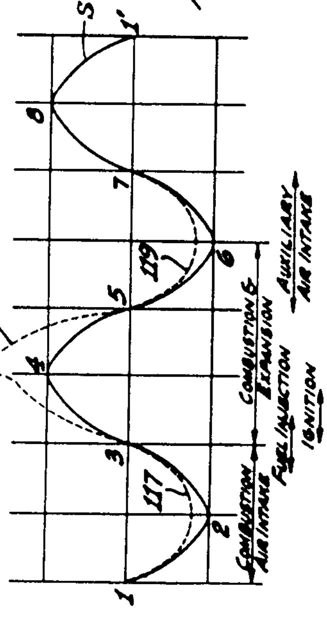

FIG. 8.

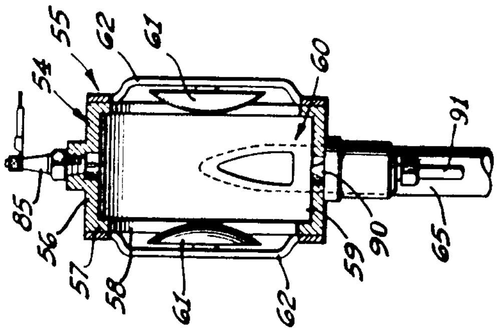

FIG. 7.

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RESONANT GAS COMPRESSOR AND METHOD

Filed April 23, 1945

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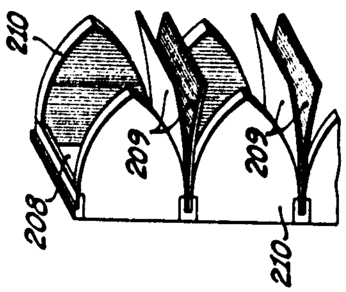

FIG. 15.

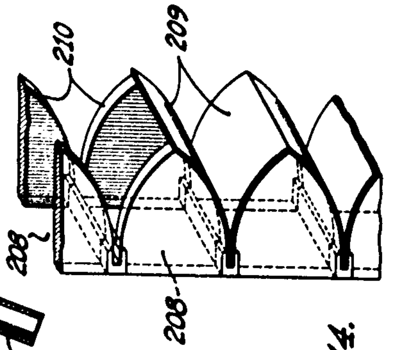

FIG. 14.

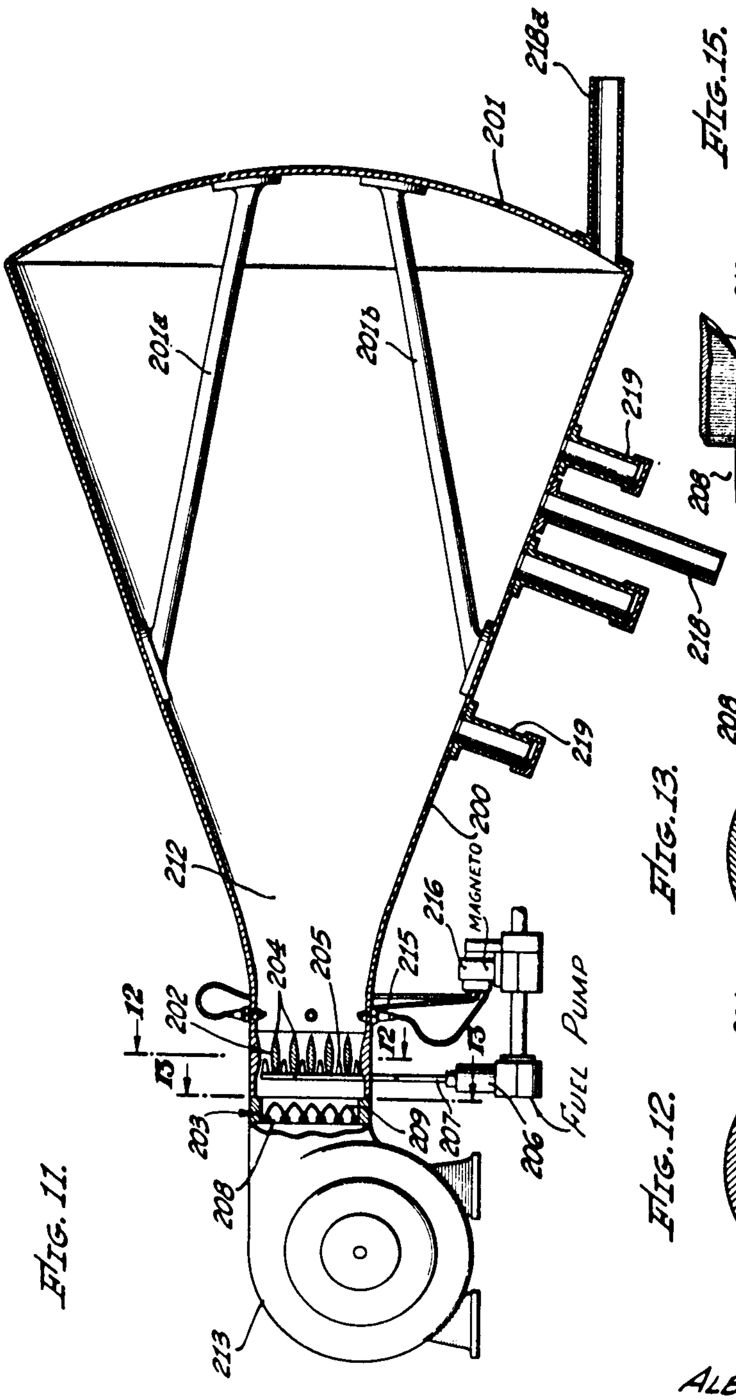

FIG. 11.

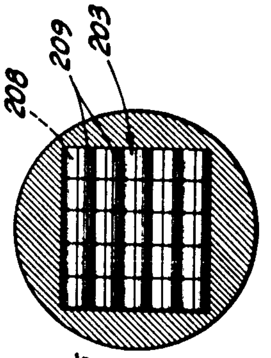

FIG. 13.

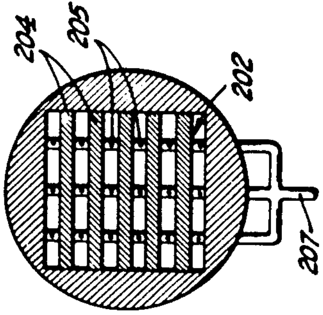

FIG. 12.

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RESONANT GAS COMPRESSOR AND METHOD

Filed April 23, 1945 6 Sheets-Sheet 6

FIG. 16.

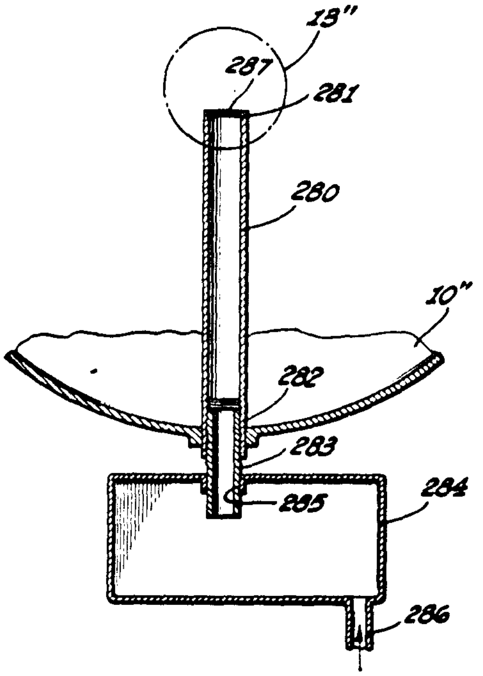
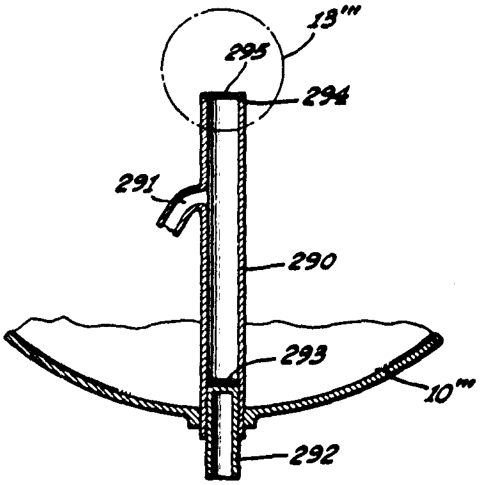


FIG. 17.



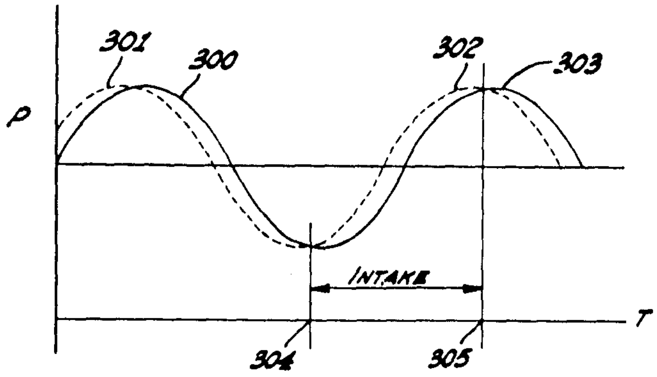


FIG. 18.

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UNITED STATES PATENT OFFICE

2,581,902

RESONANT GAS COMPRESSOR AND METHOD

Albert G. Bodine, Jr., Van Nuys, Calif.

Application April 23, 1945, Serial No. 589,753

24 Claims. (Cl. 60—35.6)

This invention relates to a novel method and apparatus for compressing or pressuring gases for various purposes; also, to a novel cavity resonator employing sonic principles and capable of producing extreme pressure variations and, if desired, a relatively-constant-pressure stream of gas, in some instances comprising products of combustion. The invention has numerous uses and the title hereinemployed is not to be considered as limiting but, rather, exemplary of one use of the device. Pressured gas from a resonating cavity may advantageously be used for purposes of jet propulsion, or to drive a gas turbine, or for pumping a liquid, etc.

In general, the invention employs sound waves in a fluid-filled cavity to establish therein pressure and velocity undulations varying in intensity in different portions of the fluid. If a pressure disturbance is caused within any cavity, certain frequencies, said to be resonant, tend to be amplified and to establish a standing wave pattern throughout the fluid, while all other frequencies tend to be damped out. The certain frequencies which tend to maintain themselves are determined by the shape and dimensions of the particular cavity. If pressure pulses are repeated at a resonant frequency, preferably in a region of maximum pressure variation, the resonating wave tends to increase until a condition of equilibrium is reached at which the input energy equals that dissipated in the form of frictional losses.

Such a system may be used as a pump or compressor by introducing new fluid into the chamber during a period of rarefaction at the point of intake. Discharge may be either continuous or only during periods of a selected recurring phase and may be located at any point within the cavity. The pressure pulses are preferably generated by the intermittent combustion of fuel, the hot gases produced thereby being permitted to expand and do work at a preferred region before dispersion, all within the resonating cavity.

In my copending application entitled Resonant Wave Pulse Engine and Process, filed November 3, 1947, Serial No. 783,662 and now issued as Patent Number 2,480,626, which application was a continuation-in-part of my original application filed April 21, 1942, for Method and Apparatus for Generating a Controlled Thrust, Serial No. 439,926, the latter now abandoned, a resonating cavity was disclosed consisting of a pipe closed at one end and opening at the other into a capacitance chamber. The sound waves are generated within the closed pipe end so as to produce a plane wave normal to the axis of the pipe. When this wave emerges into the capacitance chamber from the opposite end of the pipe it diverges and the wave front changes from a plane to a curved surface. It is to be noted that in this and other previously proposed applications utilizing resonating combustion chambers, an important portion of the system has generally been designed for one dimensional vibration. It is a novel feature of the present invention that the standing wave diverges immediately from the generator, in certain versions in all directions. Furthermore, in most embodiments, a generator which preferably functions substantially as a spherical-wave point-source is used whereas in previous devices utilizing resonating gas systems, it was usually desirable to produce a wave as nearly planar as possible, particularly in that portion adjacent the source. In fact, one identifying feature of long, small cross-section pipes is that they tend to generate plane waves by multiple wall reflection regardless of generated wave form. Helmholtz resonators are in the same analogous class as pipes because they also deal with one dimensional motion, even though they must be short in all directions relative to a quarter wave length.

It is an object of this invention to provide a novel combustion resonator for any purpose and in which the wave front is not everywhere a plane of substantially constant area (non-planar), e. g., to provide a sonic resonator in which the desired resonant wave radiates divergently from the generator. Such an acoustic system tends to set up a standing wave having a pressure antinode of maximum energy density in the region of the generator. This feature makes possible a highly efficient thermal excitation since the effective compression-expansion-ratio in the zone of intermittent combustion is very high.

In a tube-confined column of fluid subjected to an energizing disturbance, a condition of standing wave resonance establishes in the column one or more zones of large or maximum pressure variation, in which zone or zones the velocity variations are a minimum. Spaced one quarter wave length from any such zone of maximum pressure variation, or spaced midway between two such zones, is a zone of large or maximum velocity variation. In other words, the energy in a resonating column changes in form (but not in quantity) from section to section along the length thereof, being primarily potential or pressure energy at the zone of large or maximum pressure variation and primarily kinetic or velocity energy at zones of large or maximum velocity variation. In the latter zones, the gas molecules vibrate parallel to the axis of the column. In high-energy sonic systems, the oscillation of such gas molecules in or near a zone of maximum velocity variation creates subtantial friction at the inner surface of the confining tube and, at extremely high velocities, shock waves are created, tending to absorb energy. These factors tend to limit the power which can be generated by the sonic vibration in such a column, and tend in some instances to interfere with the desired wave pattern.

It is accordingly an object of the present invention to establish a zone of large or maximum velocity variation in space with a minimum of contact with the fluid confining surface.

By way of example, if a sound wave generator is disposed in the center of a hollow fluid-filled sphere and if the frequency is such as to produce sound waves having a wave length equal to 1.4 times the radius, a condition of standing wave resonance will be maintained within the sphere, as may be ascertained, for example, by reference to any edition of Lord Rayleigh's Treatise on Sound. Zones of pressure loops or antinodes will exist at the center of the sphere and adjacent the inner surface of the spherical wall. For the condition of resonace at fundamental frequency, the zone of large or maximum periodic velocity is revealed by Lord Rayleigh's equations to be located approximately a half wave length from the center, that is, in an envelope about two thirds the radial distance from the center to the spherical wall. In this latter spherical zone, the gas molecules will oscillate along paths radiating from the center of the sphere, but the only friction will be between the gas molecules themselves, and substantially no shock waves will be established.

It is to be understood that the term "wave length" as used herein always means the quotient of velocity divided by frequency. When plane waves are propagated in one direction only (i. e., without divergence) the distance between successive particles of the medium which are in phase with one another corresponds to this quotient. Particles 180° out of phase with one another are half this "wave length" apart in a one-dimensional vibrating system. The term ceases to have this significance when applied to waves propagated in more than one direction, however. If waves diverge, particles in phase with one another are separated by a distance which is greater than the "one-dimensional" wave length. If, for instance, a point source generates sound waves in a medium which propagates the waves equally well in all directions, then the loci of all particles in phase with the source is a series of concentric spherical surfaces having the source at their common center. The distance from the point source to the first spherical surface is substantially greater than the distance from the first spherical surface to the second; the distance between successive spherical surfaces decreases and as the spheres become large, and plane wave conditions are approached at high overtones, the separating distance approaching a "one-dimensional" wave length as a limit.

In many of its preferred embodiments, the invention employs cavities which have the peculiar property that pressure waves originating therewithin and reflected by the bounding surfaces thereof are concentrated in a focal zone, usually a point or a line rather than a surface. A spherical cavity presents the simplest case in which all reflected waves converge to a focal point. An important advantage of the spherical resonating chamber results from the fact that a sound wave reflected in a converging wave from the concave spherical surfaces increases progressively in energy density and therefore in the degree of pressure variation as it approaches the focal point at the center of the sphere. Pressure variations existing at the reflecting surfaces are thus enormously amplified as they approach the center. In standing wave systems using one dimensional vibration only the sound waves are generated and transmitted in a plane. The maximum pressure attained at any pressure antinode is no greater than that periodically exerted on all reflecting surfaces normal to the wave (except for attenuation). In a spherical standing wave system, on the other hand, the maximum pressures attained in the region of the centrally located pressure antinode are many times those exerted against the reflecting surfaces of the cavity. A conical sector of a sphere, having a solid angle at its vertex of any desired magnitude less than 4π, has similar properties.

Another example, comprehended by the present invention, involves a substantially cylindrical resonator having an elongated or cylindrical focal zone coaxial with the cylinder. If an elongated sound-wave generator is disposed in or adjacent this focal zone, sound waves will move radially outward from the focal zone with substantially cylindrical wave fronts and will be reflected by the inner surface of the cylindrical wall to concentrate in the focal zone. Convergence from the reflecting surface has an amplification effect similar to but somewhat less than in the case of the sphere.

The invention also comprehends other shapes of resonant cavities in which amplification of a reflected wave is achieved by means of surfaces disposed about a focal zone; also resonant cavities in which sound generators are distributed at points other than that of maximum pressure variation. Less symmetrical cavities have more complicated standing wave patterns but if they are such as to have one or more foci, an ellipsoidal cavity for example, then the establishment of a standing wave in the contained fluid will create pressure antinodes at said foci and the pressure variations at these points will exceed those occurring in any other region of the cavity.

The invention is not limited to focusing surfaces, however, but comprehends broadly resonating cavities of whatever shape, in which sound waves are radiated divergently from the generator and in which the standing wave created thereby is utilized for the performance of mechanical work, such as pumping fluids. Even in the absence of points within the cavity which constitute foci, if vibration occurs in more than one direction certain pressure antinode regions will be characterized by a wider variation in pressure than others. Preferably, however, the sound waves are generated adjacent a focal zone, or in the absence of a focal zone, near a pressure antinode of maximum strength in order that reflected sound waves will reinforce the undulations of the sound-wave generator.

Certain later-described peculiar advantages of resonating cavities which have one or more pressure antinode regions of maximum energy density may be explained in terms commonly used with reference to vibrating systems whether they be electrical, mechanical, or acoustical, and for convenience these will here be defined. A dimensionless figure of merit represented by the letter "Q" is used to express the ratio of energy stored in a vibrating system to the energy dissipated by it during each half cycle. Since energy is stored only to the extent that resonance exists, Q reaches a maximum at each resonant frequency and declines in value with departure to higher and lower frequencies. The term acoustic impedance Z, represents the ratio of alternating pressure component to the volume velocity, the latter being the integral of particle velocity over the area of a surface past which it occurs. In a standing wave system the impedance quotient has a maxima in the pressure antinode regions since an added increment in the applied alternating pressure results in very little increased velocity at this antinode. The terms "Q" and "Z" are related in that a system having a high Q will exhibit an impedance at its pressure antinodes which is relatively large in comparison with the impedance at its velocity antinodes. In fact, as attenuation approaches zero, "Q" and the pressure antinode impedance approach infinity and the impedance at the velocity antinode approaches zero.

When a standing wave in a fluid is caused to perform mechanical work it is damped or attenuated, such as by the substraction of vibrating fluid and its replacement by static fluid which must be compressed, or accelerated, depending on whether intake is near a pressure or velocity antinode region. In order that the standing wave may continue to exist, new energy must be added, generally once in each cycle, and, in the preferred embodiments of this invention, this is accomplished by the intermittent combustion and expansion of fuel and air. In such a system, it is desirable that the stored energy of the standing wave be large in proportion to the energy lost in the pumping of fluid and in normal frictional attenuation. The standing wave must carry a sufficient reservoir of stored energy to compress each newly introduced charge of fluid despite these losses. It is seen that the standing wave performs functions exactly analogous to those of a flywheel, the energy being stored for part of the cycle in the fluid mass at the velocity antinode. The two energy storage device differ from one another in that the flywheel stores its energy wholly in the form of kinetic energy whereas the standing wave alternately stores energy as kinetic energy at the velocity antinodes and as potential energy of elastic deformation at the pressure antinode. They resemble each other, however, in that for a given frequency, the capacity for energy storage goes up with the magnitude of the moving mass. In a spherical cavity, for example, the moving fluid mass is in the shape of a thick walled hollow sphere whereas in the ordinary one-dimensional resonator it is in the shape of a cylinder or prism. It is seen that for an apparatus of a given bulk and for a given size of combustion chamber, the sphere has a much heavier "flywheel," and, therefore, one which provides a greater energy reservoir at a given alternating velocity. Furthermore, the wall drag encountered in a one-dimensional resonator is totally absent in a sphere, and in other diverging wave cavities is substantially less for a given mass of moving fluid. Such cavity resonators, therefore, provide a flywheel of considerably less friction. Stating exactly the same point in a slightly different manner, a cavity resonator, particularly a sphere, has a higher Q than a one dimensional resonator of the same bulk since attenuation is reduced by the elimination or reduction of wall drag and by reduction of internal friction since kinetic energy is stored by a large mass moving at a relatively low velocity rather than by a small mass moving at a high velocity. Partly because of this higher Q and partly because of the increased energy density (decreased wave front area) at the focal zone antinode, pressure variation amplitudes attainable there are much higher than could be attained in a one-dimensional resonator of similar bulk and with the same exciting force; for example, the focal zone pressure antinode of a sphere has a higher acoustic impedance than any other type of resonator, other things being equal. If the sound wave is generated by the intermittent combustion of fuel in this region of high impedance, then the advantages of a high compression engine are attained since high pressure variations occur there. In effect the high impedance provides the expanding gases with a loading which more nearly approaches the pressure which may be exerted by a gas expanding in an isentropic manner. Low impedance systems permit free expansions to a large extent, with resulting increases in entropy and losses in thermodynamic efficiency.

In high-energy sonic systems, it is often desirable to employ intermittent combustion to establish the sound waves. The present invention has among its objects the burning of increments of fuel in a sonic cavity to establish and maintain the desired sound wave pattern in the cavity. Another object of the invention is to burn increments of fuel at a position near the focal zone of a reflecting surface in such a manner as to establish a condition of standing wave resonance giving rise to pressure variations at the focal zone which, if desired, can be extremely large. Other objects reside in the utilization of the sound waves in a sonic cavity to aid in the intake of air and fuel, the compression and ignition of the fuel-air mixture, the expansion of the fuel, and/or the exhaust of gases, comprising products of combustion, from the sonic cavity. One of the general objects of the invention is the provision of a novel internal-combustion resonator useful for many purposes.

Other objects of the invention reside in the utilization of sound waves or sonic energy in the compression of fluids, particularly gases. Thus, it is an object of the invention to provide a gas-pumping or compression system in which the pressure-increasing means is combined with a storage vessel; also, to provide a simple, highly efficient, internal-combustion compressor. In this latter connection, it is an object of the invention to burn increments of fuel, mixed with appropriate amounts of air, at a position spaced from reflecting surfaces in such manner as to utilize a reflected wave to produce large variations in pressure at a focal zone and to remove compressed gas, usually comprising products of combustion, from a position removed from the point of fuel burning, whereby the gases at the point of withdrawal are not subject to the large pressure variations in the focal zone.

Certain general objects of the invention include the provision of a novel method and apparatus for maintaining large pressure variations within a container without transmitting equal pressure variations to the walls of the container; provision for the introduction of relatively-low-pressure gases to a sonic cavity and for the discharge of relatively-high pressure gases therefrom; provision for intermittent intake of gases to, and a relatively constant-flow discharge of gases from, a sonic cavity; provision of a resonating system including a sonic cavity in which increments of fuel can be burned at a natural resonant frequency of the sonic cavity or at a frequency which is a multiple or sub-multiple of its resonant frequency; provision for the timed intake of fuel and combustion-supporting air to a sonic cavity and, if desired, a supply of auxiliary air thereto; provision for the confinement of fuel-burning to a small zone, usually near the focal zone of a curved reflecting surface; employment of the burning of increments of fuel for the development of pressure to discharge a relatively-constant-pressure stream of the products of combustion diluted, if desired, by another gas such as air; provision for the maintenance of a high mean pressure in the sonic cavity, conducive to desirable wave patterns and high-energy outputs; provision for a novel control of valves in a sonic system; and provision of an automatic control to compensate for differences in load or amount of gas withdrawn. It is an object of this invention to provide a sonic engine which utilizes to full advantage the natural spherical expansion pattern of explosions.

Other objects of the invention reside in the provision of a system for turning a rotary shaft in step with the sound waves in the sonic cavity, i. e., at a speed proportional to the sound-wave frequency; the turning of such a rotary shaft by a turbine receiving a pressured stream of gas from the sonic cavity; and the utilization of at least a portion of the power transmitted by such a shaft for driving a supercharger or compressor for forcing air into the sonic cavity, or for timing the intermittent introduction of fuel to, or the burning of fuel in, such a sonic cavity. It is also an object of the invention to conduct any desired portion of the gases from a sonic cavity of the internal-combustion type for use in driving a turbine, any desired portion of the power developed by the turbine being used to drive a compressor aiding in the forcing of gases into the sonic cavity, any remaining portion of the power developed by the compressor being used for any desired purpose, and any residual energy in the gases discharging from the turbine being used for any desired purpose, for example by discharge as a propulsive jet. From this standpoint, the invention is not limited to a governing of the turbine speed to be proportional to the pressure undulations in the sonic cavity or the explosion frequency therein.

Further objects and advantages of the invention will be evident to those skilled in the art from the following description of an exemplary embodiment, namely, an internal-combustion cavity resonator employed as a compressor.

Referring to the drawings:

Figure 1 is a vertical cross-sectional view of a simple form of internal-combustion spherical resonator employed as a compressor;

Figure 2 is a fragmentary vertical sectional view of an alternative air and fuel intake system useful in the system shown in Figure 1;

Figure 3 includes graphical representations of pressure and velocity variations in the sphere of Figure 1;

Figure 3*a* is a detail sectional view on an enlarged scale of the fuel pressure regulator;

Figure 4 is a diagram illustrating the timing relationships of an internal-combustion resonator operating as a "two cycle" system;

Figure 5 is a vertical sectional view of a spherical resonator employing novel valve and combustion means;

Figure 6 is a vertical sectional view of an alternative form of spherical resonator;

Figure 7 is a vertical sectional view of the combustion chamber in the center of the sphere of Figure 6, taken along the line 7—7 thereof;

Figure 8 is a diagram illustrating the timing relationships of an internal-combustion resonator operating as a "four-cycle" system;

Figure 9 is a fragmentary view similar to Figure 7 and showing an alternative form of combustion chamber;

Figure 10 is a vertical sectional view of an alternative internal-combustion resonator operating as a compressor and employing a cylindrical cavity;

Figure 11 is a vertical sectional view of a conical sonic cavity which can be employed instead of the spherical or cylindrical forms;

Figure 12 is a vertical transverse section taken through the cone of Figure 11 along line 12—12 in the direction of the arrows;

Fig. 13 is another vertical transverse section through the cone on plane 13—13 of Figure 11;

Figures 14 and 15 show a set of reed valves in open and closed position, respectively;

Figure 16 is a sectional view of a pipe and apparatus for supplying combustible mixture to a resonating chamber;

Figure 17 is a sectional view of another intake pipe means; and

Figure 18 is a graph of the pressure cycle of a fuel intake opening of the type shown in Fig. 16 or Fig. 17.

Referring particularly to Figure 1, wherein is shown a simplified spherical resonator or compressor, the apparatus includes a hollow sphere **10** defining a closed sonic cavity **11**, the hollow sphere providing a curved reflecting surface **12** which, in this embodiment, is substantially spherical. The curvature of this surface **12** is such as to reflect sound waves towards, and concentrate them in, a focal zone in or at the center of the sphere, the focal zone in this instance being substantially spherical and being suggested by the dashed line **13**. If the curved surface **12** is truly spherical and if sound waves are generated in the center of the sphere, the sound waves reflected by the curved surface **12** will concentrate at a focal point exactly in the center of the sphere. The sphere **10** may be supported by any suitable means, such as supports **14**.

A sound-wave generator **15** is disposed within or adjacent the focal zone **13**. The preferred sound-wave generator comprises means for intermittently burning increments of fuel to create pressure pulses. It will be clear, however, that various types of sound generators can be employed so long as they transmit pressure pulses to the fluid in the sonic cavity **11**.

As shown, the device includes a carburetor **16** for producing a combustible fuel-air mixture, the air being supplied through an air duct **17** and the fuel being supplied through a fuel line **18**. This carburetor may be of conventional design and may have the usual throttle **19**. A blower, supercharger, or compressor **20** increases the pressure of the fuel-air mixture and forces it along a pipe **21** into the focal zone **13**. The compressor **20** may be driven at a speed proportional or non-proportional to the frequency of the sound-wave generator **15** and this drive may be from any suitable means, for example from a turbine **22** interconnected with the compressor by a shaft indicated by dashed line **22***a*.

An inlet check valve is preferably provided at some position along the pipe **21**, being shown as including a poppet valve **23** urged toward closed position by a spring **24**; it has been found, however, that the apparatus will function if no valve is provided, the combustible mixture being admitted in "gulps" because of the pressure variations in the standing wave. By the same token, valve **23** can be provided with sufficient inertia and damping so that it will remain open while the system is operating, and will automatically close as a check valve to retain storage of static pressure when the system is not operating. An ignition device, such as a spark plug **25**, is positioned adjacent or within the focal zone **13**, being here shown as carried by the inner end of pipe **21**; and this ignition device may be energized through lead **26** extending through insulator **27** to a buzzer spark coil, step-up transformer, or other energizing source, indicated by the number **28**. Closing an ignition switch **29** establishes a spark in the gap of the spark plug **25**. The spark may be used continuously if desired, but in most embodiments the spark is required for starting only, since under operating conditions there is usually a small amount of continuous combustion providing a "tail-flame" for igniting each successive charge.

A discharge pipe **30** extends through the wall of the sphere **10**. The inner end of this discharge pipe terminates preferably in a zone of the cavity **11** in which pressure variations are smaller than in the focal zone **13** or adjacent the reflecting surface **11**. In the particular system illustrated, the wave length of the sound waves for fundamental resonance is equal to 1.4 times the radius of the sphere and, in this circumstance, the inner end of the discharge pipe **30** preferably terminates approximately two-thirds the way from the center of the sphere to the reflecting surface **12**, this being in the locus of a pressure node (velocity anti-node) of the system, so that a substantially constant pressure discharge is achieved. The innermost end of the discharge pipe may openly communicate with the fluid in the sonic cavity **11** but, in some instances, I prefer that this communication be through a damping screen, **32**, thereby tending to prevent resonance in the discharge pipe **30**. The discharged gases move through a main valve **33** and thence through a valve **34** to be utilized for any desired purpose, such as driving air drills, pneumatic machinery, etc. Any desired portion of the discharged gases may move through valve **35** to the turbine **22** for driving same, the power from the turbine being used at least in part in driving the compressor **20** any residual power from the turbine being available for driving auxiliary equipment or for any desired purpose. The gases discharging from the turbine **22** may also be used for any desired purpose and are shown as discharging through nozzle **36**. It is often desirable to provide a spring-loaded discharge valve at some portion of the system. Such a valve is suggested by the number **37** as connected to the discharge pipe **30**, and opens to prevent building up of destructive pressures in the sonic cavity **11**. Also, if it is desired to obtain gas at pressure higher than the mean pressure of the cavity then valve **33** may be closed and the spring of valve **37** adjusted to open only during periods of maximum pressure or valve **37** may be highly damped and not responsive to each momentary pressure but serving only to restrict discharge and permit the establishment of high mean pressure within the cavity **11**. The highly compressed gas may be carried away through pipe **37***a*.

The operation of the simplified embodiment shown in Figure 1 can best be illustrated by assuming, first, that the pipes **21** and **30** are absent and that a single minor explosion is initiated at the center of the sphere when filled with a gas, such as air. For example, if a blank cartridge or an explosive cap were detonated at the center of the sphere, the resulting minor explosion would establish a steep wave front sound wave moving radially outward toward the reflecting surface **12** at a speed corresponding to the speed of sound in the gas filling the sonic cavity **11**. It is important to note that the wave front of such a sound wave is substantially spherical and corresponds to the curvature of the surface **12**. Correspondingly, the wave front of this sound wave will reach all portions of the curved surface **12** substantially simultaneously. Since the wave impinges normally against the surface at all points, it is reflected normally inwards. When reflected, the wave converges from all parts of the spherical surface with increasing pressure as it approaches the focal zone **13**.

The sound wave thus transmitted from the center of the sphere to the surface **12** and reflected back toward the center of the sphere can be considered as a steep wave front pressure pulse. The actual intensity of the pressure pulse when first generated in the focal zone may be, for example, several hundred pounds per square inch. However, due to the inverse square law, the actual unit pressure of the pulse decreases during outward travel so that the internal pressure on the curved surface **12** will be very substantially less than the initiating pressure. However, it is important to notice, also, that the reflected pressure pulse is concentrated and amplified as it retraverses the sonic cavity toward the focal zone because of the focusing action of the curved surface **12**. Correspondingly, the pressure in the focal zone **13** will again be built up because of the reflecting wave to a value which is lower than the initiating pressure only by the losses in wave transmission and reflection. This will cause a compression of the gas in the focal zone which, upon subsequent expansion, will again send a pressure pulse outward toward the curved surface **12**. Actually, the pressure pulse traverses and re-traverses the sonic cavity repeatedly. In fact, if a blank cartridge or cap is detonated at the center of such a sphere, the sound waves will continue for a substantial period of time, dying out only because of losses in the system.

On the other hand, it follows from elementary principles of sonics that the pressure pulse thus generated in the focal zone tends to create, upon its expansion, a rarefaction of a magnitude substantially equal (actually equal pressure values only if mean pressure is sufficient) to the pressure pulse and having a peak negative pressure one-half cycle after the peak positive pressure. Correspondingly, if the distance between the focal zone and the curved surface **12** is such that pressure anti-nodes are established at the center and at the reflecting surface, then at fundamental resonance a rarefaction exists in the focal zone at the instant the pressure pulse reaches the curved surface 12 for reflection. It also follows that, when the reflected pressure pulse again raises the pressure in the focal zone 13 to a maximum, a rarefaction will exist immediately inside the curved surface 12.

Assume, next, that the pipes 21 and 30 are in place and that a small combustible charge is forced through the valve 23 into the focal zone 13 and is ignited by the spark plug 25. A pressure pulse similar to that previously described will move outward and be reflected by the curved surface 12 and the pressure in the focal zone 13 will vary widely about a mean pressure value, increasing when a pressure pulse is present in the zone 13 and decreasing when a rarefraction is present. The system can be designed to open the valve 23 momentarily and admit an additional increment of fuel when a sufficient rarefraction exists in the focal zone 13, preferably by virtue of the occurrence of such rarefraction. In the present illustrative arrangement, the valve 23 is urged toward closed position by the pressure in the focal zone and by the spring 24, and is urged toward open position by the atmosphere plus compressor-induced pressure in the pipe 21. As the pressure in the focal zone 13 decreases, a point is reached, depending upon the adjustment of the spring 24, at which the forces holding the valve normally closed are overcome and the valve opens to admit an additional increment of combustion charge. Due to the cyclic variations in pressure in the focal zone 13, this additional increment of fuel-air mixture tends to be compressed either before or soon after ignition is initiated by the spark plug 25.

By way of analogy to the operation of a two-cycle engine, I have termed as a "two-cycle" operation the introduction and burning of a fuel charge in step with each pressure variation in the focal zone 13. The relationships are illustrated in Figure 4, in which is shown a sine wave 38 drawn about an abscissa base 39. Ordinates can be considered as representing pressure and abscissa as representing time, in which event the sine wave 38 approximately represents pressure variations in the focal zone when the system is operating under idling conditions or when it is resonating following a single explosion in the focal zone. The abscissa base 39 in this instance is a representation of mean static pressure in the sonic cavity which may be many times atmospheric pressure. Usually, I prefer to employ the invention with a mean pressure above atmospheric pressure and, in high output systems, more than twice atmospheric pressure, a high mean pressure having advantages described elsewhere herein.

In Figure 4, one complete cycle is represented between points 1 and 5, a rarefraction being present between points 1 and 3 and a condensation or pressure pulse being present between points 3 and 5. The reduced pressure between points 1 and 2 is sufficient to open the valve 23 to admit an additional increment of fuel-air mixture. Preferably, the intake period is very short, as suggested in Figure 4. It is followed by compression and ignition between points 2 and 4 and, in effect, by expansion between points 4 and 5 and into the next cycle, as suggested in Figure 4. The burning of the fuel is not instantaneous since it depends upon the rate of flame propagation but it does occur so rapidly that the pressure curve tends to rise sharply. In practice, a graphical representation of the actual pressure in the focal zone will not follow a true sine wave for two reasons. In the first place, intake between points 1 and 2 will, in itself, tend to reduce the existing rarefraction so that the curve follows more nearly the dotted line curve 41 during this period of the cycle and, in the second place, the burning of the additional increment of fuel tends to increase and sharpen the positive pressure peak 4 so that the curve approaches more nearly the dotted curve 42 in the compression portion of the cycle. However, this distortion of the sine wave (even in filtered systems with one degree of freedom) from impulse excitation and irregular combustion rates is far less radical with the three dimensional resonators of this invention as compared with pipes, principally because of better Q loading. Also, with spherical resonance the harmonic frequencies are in a proportion of substantially 3, 5, 7, etc. This means that periodic combustion controlled for one frequency is less apt to excite overtones than in resonators such as pipes with harmonic series of 1, 2, 3, 4, etc.

It is important to understand that, if resonant conditions are established in the sonic cavity 11, the peak pressures in the focal zone 13 can be built up to a value many times higher than the peak pressure of a single explosion. This is true because of the automatic timing of subsequent explosions to reinforce the pressure peaks existing because of reflection of previous pulses. Thus, even minor explosions in the focal zone can be made to reinforce resonantly the pressure undulations therein existing, thereby building up enormous pressures in this zone. Further, such momentary pressures exist, in effect, in space and need not be confined by extremely massive walls. The invention is capable of producing very large pressure variations in the focal zone 13, while pressures substantially less than 100 lbs./sq. in. are applied to the inner wall of the sphere.

If resonant conditions are established in the sonic cavity 11, the focal zone 13 will be a zone of large or maximum pressure variation, indicated by the letter P in Figure 3. Likewise, the zone immediately inside the reflecting surface 12 will be another zone of maximum pressure variation, indicated by the letter P', though the pressure variations at P' will be substantially less than at P. At spherical zone V, approximately two-thirds the way from P to P', pressure variations will be a minimum and the sonic energy will be primarily kinetic or velocity energy. Zone V is a zone of large or maximum velocity variation and small or minimum pressure variation. The relationship of pressure and velocity is shown in the curves of Figure 3, in which the abscissa scale represents the radius of the sphere, the lines A—A and B—B respectively corresponding to the center of the sphere and the side wall or reflecting surface 12. Ordinates of the upper curve represent pressure and the envelope between the two curves represents pressure variations at different radii. It will be noted that the pressure is a minimum at zone V. In the lower curve, ordinates represent velocity, and it will be noted that the envelope of the curves shows maximum velocity at zone V. This is the fundamental frequency standing wave pattern. I find the fundamental frequency to have an advantage over the higher harmonics because it simplifies the problem of automatic valves.

These relationships exist for a sphere resonating at its natural frequency, i. e., a sphere having a radius equal to .715 times a wave length. However, the invention contemplates that the sphere may be made to resonate at overtone frequencies as the wave length is reduced relative to the radius of the sphere. If the sphere is resonated at its natural frequency the discharge pipe 30 preferably communicates with the zone V, whereby a substantially-constant-pressure stream of gas comprising the products of combustion can be withdrawn.

If the sphere is resonating at an overtone frequency, there will be two or more zones V between the center and periphery of the sphere and the inner end of the discharge pipe will in such event preferably be positioned to open on one of these. In other instances, however, pressure fluctuations in the discharge are not disadvantageous and, in some instances, these are desirable, particularly when valve 31 is adjusted for periodic opening at peak pressures for discharge of elevated pressures, or tuned inductively or capacitatively for discharge at other chosen phase relationship. Broadly, therefore, the inner end of the discharge pipe 30 can communicate with the sonic cavity 11 at any desired position. If, for instance, the discharge is located at the inner surface of the sphere, the discharged fluid is subject to minor pressure variations since the bounding surface constitutes a pressure node. The alternating wave energy is, however, at a minimum at this surface; fluid may therefore be abstracted under proper conditions with small effect on the standing wave within the cavity.

Although most of the embodiments herein disclosed provide for the intermittent introduction of fuel or air, or both, it is also possible to initiate and maintain resonance with fuel and air which is supplied continuously if the combustible mixture is introduced into a region of the cavity which is subject to large pressure variations under conditions of resonance. When constant combustion occurs in such a region, an unstable equilibrium exists; the slightest variation in the rate of combustion initiates a pressure pulse which tends to be succeeded by local alternations in pressure which increase in amplitude until a condition of equilibrium is reached. During each compression period the inflow velocity of fuel and/or air is momentarily suppressed by the increase in back pressure and combustion rates are momentarily further accelerated by the increase in the density and pressure of the burning gases. These two effects thus combine to cause a period of maximum combustion during the compression period followed by a substantial decline in combustion as the rarefaction period begins. During the rarefaction period, the lowered local pressure permits high velocity inflow of fuel and air which may be faster than flame propagation, thus combustion is not at a high rate until after a period of some ignition delay which the next succeeding compression period again increases local pressures and densities. The occurrence of combustion during the compression cycle under conditions wherein combustibles are introduced during the preceding rarefactions relies in part upon ignition lag or delay which latter can be aided by the chemical properties of the fuel and temperature and uniformity of incoming fuel-air mixture. Under these operating conditions, combustion continues during rarefaction periods but at a rate which is extremely small in comparison with that occurring during compression periods. This continuing combustion, however, does make it possible to dispense with any ignition system except for purposes of starting.

In some instances, it is desirable to supply the air and fuel separately to the sonic cavity, and this can be accomplished by the system shown in Figure 2, in which no carburetor 18 is employed, the compressor or supercharger 20 merely forcing a stream of air through the pipe 21 and the valve 23. The fuel is injected in time relationship with the wave, e. g., during the intake period indicated in Figure 4, properly by a suitable injection pump forcing an increment of fuel through a fuel line 44 and an injection nozzle 45. The fuel ignites either by pressure ignition or, if desired, by impingement on a "hot spot" provided, for example, by a member 46 supported on arm 47 to lie in the path of the injected fuel. The surface 12 is preferably smoothly polished or plated in order that heat radiated from the focal zone during combustion will be reflected back to the focal zone, thereby increasing the efficiency of thermal excitation.

Figure 5 illustrates a resonating sphere 10' similar to that shown in Fig. 1 but having two coaxial pipes 251 and 252 which enter the sphere 10' from opposite sides which may intermittently deliver a combustible charge into the central focal zone indicated by dashed line 13'. A spark plug 25' and auxiliaries are shown and may be used either continually or for starting only, as previously explained in connection with Figure 1. Combustion is found to be more effective in resonating the sphere 10' if the charges of gas and air issuing from pipes 251 and 252 are directed into open ended cylindrical shroud 253, the shroud being supported coaxially with the pipes by means of metal straps 255.

A fire screen 254 is transversely positioned in the end of each of the pipes 251 and 252 so as to prevent flames from the cavity entering the pipes and causing premature combustion therein. Such a screen was previously shown in my copending application entitled "Method for Generating a Controlled Thrust" filed April 27, 1942, and numbered 439,926.

Each of the pipes 251 and 252 is supplied with a fuel and air mixture through identical apparatus. Pipe 251, for example, opens at the opposite end to the atmosphere and is provided at some convenient point along its length with a reed type check valve indicated generally by the arrow 256 and comprising a resilient metal flap 257 attached at its upper edge to an internal annular shoulder 258 disposed in a plane oblique to the axis of pipe 251. When a rarefaction occurs at focal zone 13' and causes the pressure within sphere 10' to fall below atmospheric pressure, this rarefaction is transmitted through pipe 251 so that flap 257 is deflected to a position indicated by the dotted lines 259 and a charge of air is inducted through 251 into the sphere.

Gaseous fuel, supplied through a pipe 260, is passed through pressure regulator 263 to fuel pipe 261 at a rate which is controlled automatically by air pressure in air intake pipe 265. Pressure regulator 263 is separated into upper and lower chambers 263*a* and 263*b* by a diaphragm 264 to the lower side of which is attached a hook shaped element 262. The point of this element fits into the opening in fuel supply pipe 260. The upper chamber 263*a* communicates with air intake pipe 265 so that the gas entrance pressure to fuel orifice 267 will always be related to air entrance pressure to air orifice 256*a* because the pressure in 263*a* influences the pressure maintained in 263*b* by the diaphragm controlling the amount of fuel released by regulator valve

262 in a manner common to all pressure regulators. Fuel-air mixture ratio may be manually controlled by adjusting the opening of orifice **267** by means of needle valve **268**. Air intake pipe **265** and gaseous fuel intake pipe **261** merge into a single pipe **269** which in turn enters pipe **251** at some point between check valve **256** and the opening of pipe **251** into focal zone **13'**. Air intake pipe **265** and fuel intake pipe **261** are provided with reed type check valves **270** and **271**, respectively, which are constructed in the same manner as check valve **256**, and like it, open to permit the induction of fuel and air during the occurrence of rarefaction in focal zone **13'**. Since both air orifice **265a** and fuel orifice **267** discharge into the same negative pressure transmitted from zone **13**, the fuel-air ratio will remain stable no matter what may be the degree of pressure reduction at zone **13** because the mixture is controlled by the ratio of orifices **267** and **265a** and the equivalent introduction pressure established by regulator **263**. If desired, one or more pipes for excess air may be provided in addition to the pipes **251** and **252** which deliver a combustible mixture. One such pipe **272** is illustrated and is provided with a reed type check valve **273**.

The various check valves **273**, **256**, **270**, and **271**, may be located at various distances from focal zone **13'** in order that their openings may occur in any desired sequence. In the arrangement illustrated, the excess air valve **273**, is nearest focal zone **13'** and will, therefore, open to admit fuel-free air for purposes of scavenging before the opening of the other valves. Fuel valve **271** is positioned most remotely from focal zone **13'** in order that it may open last during each rarefaction period because of wave transit time from zone **13**, thereby delaying the entrance of fuel until after substantial scavenging has occurred.

A throttle **274** controls the intake of air through pipe **265** and, indirectly, by means of pressure regulator **263**, and orifice **267**, the amount of gaseous fuel admitted through pipe **261**.

In some embodiments, it is convenient to provide a plurality of reed type intake valves **275** at the surface of the sphere. In this way it is possible to pump large volumes of air at low pressure differential. These valves are remote from destructive flame temperatures; and therefore can be of many different materials, mounted stressed, as reeds, or relaxed to be forced-driven by the wave pulses.

As previously noted, it is usually desirable that a discharge from the resonating cavity occur with a minimum of pressure fluctuation in the efflux. In this particular embodiment the discharge pipe **276** withdraws fluid from a region of minimum energy density near the inner surface of sphere **10'**. Discharge pipe **276** is specially shaped and dimensioned to function as an acoustic band stop filter of the wave length at which sphere **10'** is resonating. Thus pipe **276** is approximately a half wave length long and has in a plane normal to its axis at its mid-point (i. e., λ/4 from each end) a short section in the shape of a cylinder having a radius of such dimensions that it reflects band-stop waves 180° out of phase.

A more complete embodiment of the invention, employing mechanically-timed intake and ignition, is shown in Figures 6 and 7. As there shown, a hollow sphere **50** provides a sonic cavity **51** and a curved reflecting surface **52** having a central focal zone **53** and being supported by suitable supports, all as previously described. Here again, the sound waves are established by combustion of increments of fuel. It is desirable in many instances that the explosion frequency be less than, e. g., a sub-multiple of, the natural frequency of oscillation (resonant frequency) of the sonic cavity. Such a mode of operation may have enormous advantages, among which is the fact that a small sphere can be used as the resonator without unduly frequent combustion or unduly frequent operation of the fuel injection or control apparatus. Furthermore, such a mode of operation can operate on a somewhat different cycle, for example, the "four-cycle" operation to be described with reference to Figure 6. The resonant frequency of a "two-cycle" simplified system, such as shown in Figure 1, is sometimes limited by the rate of fuel burning with certain fuels, the occurrence of tail flame, etc. If a modified cycle of operation is employed with reinforcing explosions occurring at a rate lower than the rate of occurrence of the pressure pulses in the focal zone, many advantages are present, including the presence of time intervals for combustion "blowdown" scavenging between successive explosions. It is possible to obtain more complete combustion of the individual fuel increments without the occurrence of tail flame carry over into subsequent cycles, or any continued burning during the operation of the apparatus. Good thermal efficiency results from any method for burning the charge substantially in separate definite pulses, preferably in step with all or a portion of the number of recurring wave cycles.

Many advantages accrue from the substantial confinement of the burning to a zone which is relatively small compared to the volume of the sonic cavity. Thus, in a sphere, it is desirable that combustion be confined to a relatively small zone in the center of the sphere, the remaining radius of the sonic cavity representing a wave path. In many of the embodiments of this invention focal zone confinement of combustion is accomplished by introducing the reagents directly into the desired zone and providing means for rapid local combustion immediately after entry (usually substantially completely burning the then existing charge on each wave cycle). To aid in confining combustion to such a central zone, the embodiment of Figure 6 includes an orificed body **54** forming a part of a sound-wave generator **55**. This orificed body **54** is shown as including a cylindrical side wall **56** with end flanges **57** extending a short distance inward to provide orifices **58** and to provide a peripherally confined zone **59** forming a part of a combustion chamber **60** in which the fuel increments are burned. To prevent concentrated jet effects through the orifices **58**, curved baffle elements **61** may be disposed concentric with each orifice **58** to spread expanding gases into cone-like shape. Each of these curved elements **61** may be mounted on an arm **62** which, if desired, may be flexible to permit small sidewise movement of the elements **61** to transmit sound waves directly to the gas in the sonic cavity **51** around the body **54** by actual movement or vibration of the elements **61**. In this way, these movable elements serve as periodic energy transmitting diaphragms similar to the diaphragm feature for maintaining separate the products of combustion and auxiliary pumped fluid, as shown in my copending applications Serial Nos. 439,926 and 589,754, both now abandoned, a feature which I find can be used in this invention. While the sound waves generated by such an intermittent-explosion system will not have exactly spherical wave fronts, the gas-spreading action and movement of the elements **61** tend to produce a wave front which is substantially spherical, that is, a wave front sufficiently spherical to produce the results desired.

An intake pipe **65** conducts air or a fuel-air mixture to the combustion chamber **60** and discharges substantially tangentially thereinto, as best shown in Figure 6, whereby the incoming stream is whirled by deflection in the peripherally-confined zone **59** to prevent escape from the combustion chamber **60** before ignition. This tangential entry permits much higher velocity of the incoming charge without giving problems from exceeding the rate of flame propagation and dislocating the actual combustion as must be safeguarded against with unbaffled combustion zones. Disposed at some point along the intake pipe **65** is a spring-loaded check valve **67** which may operate in accordance with the previous disclosure regarding the valve **23** of Figure 1 but which is preferably mechanically actuated by a push rod **68** engaging a rocker arm **69** carrying a roller **70** engaging a cam **71** on a rotating shaft **72**. This shaft is driven at a speed proportional to the combustion or oscillation frequency by means later to be described, and extends into a gear box **74** containing step-up gears driving a shaft **75** at a higher speed. This shaft is connected in driving relationship with a blower, supercharger, or rotary compressor **76** discharging into the intake pipe **65** and intaking from an induction unit **77** equipped with a suitable throttle, such as a butterfly, and operated by throttle arm **78**. A suitable carburetor **80** feeds a fuel-air mixture to the induction unit **77**, air entering the carburetor through an induction pipe **81** and fuel entering through a fuel line **82**. If a fuel-air mixture is being supplied by the carburetor **80**, the amount thereof entering the blower **76** and subsequently delivered to the combustion chamber **60** is controlled by the position of the throttle arm **78** which, in turn, is connected to a cable **83** extending to a suitable throttle control.

The fuel-air mixture may be ignited in the combustion chamber **60** by one or more spark plugs **85** at the periphery of the combustion chamber **60** and preferably firing together. These spark plugs may be energized by means of leads **86** extending through an insulator **87** mounted in the sphere and extending to a magneto **88** driven at a speed proportional to the shaft **72** from suitable gearing in the gear box **74**. On the other hand, if the supply of fuel through fuel line **82** is shut off, the throttle arm **78** merely controls the amount of air discharged into the combustion chamber **60**. In this instance, liquid fuel may be injected into the combustion chamber **60** through a suitable fuel injector **90** acting to atomize the discharged fuel. Increments of liquid fuel are fed to the injector **90** through a pressure line **91** communicating with the discharge of a fuel injection pump **92** driven by a cam **93** on the shaft **72**. For mixture control, the displacement control of the injector pump **92** can be linked to the air flow control in well known internal combustion engine design procedure.

The main discharge of gases from the sonic cavity **51** can be through a discharge pipe **94**, the inner end of which preferably terminates either

in the region of a pressure node (velocity antinode), as has been indicated in Fig. 6, or in a region of minimum alternating wave energy density (i. e., adjacent an outer wall), as previously described. The gases thus discharged can be used for any purpose, including those described with reference to Figure 1 and including the driving of the turbine-compressor arrangements suggested. In this embodiment of the invention, the gas discharge orifice is controlled by a valve at the inner end of the pipe **94** and comprising a rotatable valve member **95** secured to a rod **96** threaded in a spider **98** and operable by a handle **99** outside the sphere. Since closing down of this valve reduces the outflow and increases the back pressure on the system, the mean pressure within the sphere may be elevated thereby. One or more suitable pressure relief valves **100** may take the place of the valve **37** in Figure 1 to relieve undue pressure in the spherical cavity or to make possible an idling of the system, assuming that the valve **95** is closed.

Discharge pipe **94** may be enlarged at some section near its outer end to provide a chamber **94*a***. Preferably the dimensions of this chamber and the length of pipe between it and the valve **95** are selected to resonate at the frequency at which the cavity is operated, or at some frequency above or below the resonant frequency of the cavity at which an impedance can be obtained at the valve **95** which result in minimum energy losses from the resonating cavity. The bulb **94*a*** also serves as a surge chamber to smooth out the discharge.

The shaft **72** is driven by a turbine **103** or other means insuring that the speed of rotation of the shaft shall be proportional to the frequency of the desired sonic oscillations in the cavity **51**. If a condition of standing wave resonance is to be maintained in this sonic cavity **51**, the speed of the shaft **72** will be proportional to the natural resonant frequency of the sonic cavity if cam-timed fuel introduction and ignition are to be used. When employing the turbine **103** to drive the shaft **72**, a portion of the compressed gases may be removed from a zone of small or minimum pressure variation in the sonic cavity through an orifice **104** into a pipe **105**, the orifice serving to prevent tendency for sonic oscillations within the pipe. This pipe feeds the substantially constant-pressure gases to the intake of the turbine **103** to drive same, the discharge from the turbine being either into the atmosphere or being used for other purposes such, for example, as forming a jet for propulsion or other purposes. For instance, the turbine discharge may be conducted to a nozzle **106** by opening a valve **107**, or, as is often desirable, a heat exchange can be arranged with the incoming air by opening a valve **108** to conduct the turbine discharge to a jacket **109** around a portion of the intake pipe **65** preparatory to discharge through nozzle **110**. To maintain the speed of the shaft **72** constant, a governor **112** is provided having a governor arm **113** connected by link **114** to a throttle arm **115** controlling the position of a throttle **116**, such as a butterfly valve positioned in the pipe **105**. The governor controls the throttle **116** in such way that the shaft **72** will run at fixed speed regardless of the load placed on the shaft **72** by various settings of the throttle arm **78** associated with the blower **76**. During starting of the device, the shaft **72** can be driven by auxiliary means at the desired speed.

The system thus far described with reference to Figure 6 can be operated on a "four-cycle" principle, the term being used as somewhat analogous to the four-cycle operation of a piston-type internal combustion engine. With this mode of operation, the frequency of explosions will be one-half of the fundamental frequency of the sphere and one explosion will occur in step with alternate pressure pulses in the focal zone. It should be understood, however, that modified timing will permit other cycles of operation in which an explosion occurs only after a predetermined number of pressure pulses, the resonating qualities of the sonic cavity allowing it to "fly-wheel" between intervening explosions. In this connection, a large amount of energy is stored in the resonating cavity 50 and this is somewhat analogous to the flywheel of a piston-type internal combustion engine.

The "four-cycle" operation can best be explained with reference to Figure 8, which is similar to Figure 4 in that a sine wave S represents pressure variations in the focal zone under low-load, idling, or "fly-wheeling" conditions. Points 1, 2, 3, 4, and 5 long this wave designate one complete sonic cycle, while points 5, 6, 7, 8, and 1′ designate a second complete sonic cycle. Together, these sonic cycles represent one complete operating cycle. The valve 67 is timed to admit combustion-supporting air at a time between points 1 and 3 of Figure 8. If desired, the valve 67 can be maintained open for the entire time between points 1 and 3, or it can be opened for any fractional length of time. As previously mentioned, fuel can be introduced along with this combustion-supporting air or it may be injected by the injection nozzle 90 near point 3. If the fuel spray hits a hot spot in the combustion chamber 60, the initiation of combustion is automatically controlled by the timing of the fuel injection. However, if a closer control of ignition is desired, this can be accomplished through energization of the spark plugs 85, this normally occurring around point 3. The combustion reinforces pressure peak 4, after which expansion toward negative wave peak 6 occurs. Thereafter, the pressure rises to a second peak 8, by wave resonance unaided by any additional explosion, and then drops to point 1′ preparatory to starting another operating cycle. An actual plot of the pressure variations in the focal zone will not be truly sinusoidal. In fact, admission of the combustion-supporting air through valve 67 will tend to make the curve between 1 and 3 follow more nearly the dotted line curve 117. Similarly, the explosion, reinforcing the otherwise-present pressure peak 4, will tend to raise the pressure more nearly to dotted line curve 118. Admission of auxiliary air between points 5 and 7, in a manner presently to be described, will make the curve approach more nearly the dotted line curve 119.

Returning to Figure 6, it is often desirable to supply a stream of auxiliary air to the sonic cavity 51 in addition to the combustion-supporting air supplied through pipe 65. A stream of auxiliary air thus introduced may blend with and dilute the products of combustion and will be discharged at higher pressure through the pipe 94 or the pipe 105 along with the products of combustion. Figure 6 shows three systems for introducing such auxiliary air, these systems being usable singly or together. In all three systems, it is usually desirable to introduce the auxiliary air into a zone of substantial pressure variations thereby avoiding the necessity of any pump for the auxiliary air—particularly when a high mean static pressure is maintained in the cavity.

The first system for introducing a stream of auxiliary air operates in timed relationship with the shaft 72. It includes an auxiliary air-intake pipe 120, terminating adjacent the focal zone 53 just outside the comubstion chamber 60. A valve 121 is disposed at some position along this pipe, this valve being shown as of the poppet type, tending to remain closed because of the action of a spring 122. The valve is operable by longitudinal movement of a push rod 123 traversing the pipe 120 and bearing against a cam 124 on the shaft 72. In a "four cycle" operation, it is usually preferable to open the valve 121 between points 5 and 7 of Figure 8, thus tending to reduce the rarefaction toward the dotted line 119, as previously mentioned. The valve can, however, be opened during each rarefaction pulse, if desired. In any event, it should be closed before the pressure moves up too high on the subsequent pressure pulse curve.

The second system for introducing a stream of auxiliary air includes an auxiliary air-intake pipe 125 providing a spring-operated valve 126 opening upon each rarefaction pulse of sufficient magnitude to overcome the biasing action exerted by springs 127 and 128 on the valve 126, taking into account pressure variations on opposite sides of the valves. The spring 127 is stronger than the spring 128, thus tending normally to hold the valve 126 closed, the double spring arrangement is desirable because it is easier to provide any degree of valve bias against the seat independent of the spring-constant which tunes with the mass of the valve.

The third system for introducing a stream of auxiliary air to the sonic cavity 51 includes an auxiliary air-inlet pipe 130 communicating with the zone P′ immediately inside the reflecting surface 52 of the sphere. A valve 131, biased toward closed position by spring 132, is employed to admit air each time a rarefaction pulse exists in the zone P′ sufficient to overcome the action of the spring.

I have found that an auxiliary air system such as the second or third above mentioned will permit continued operation even if the first mentioned system is used for mechanical control of discharge phasing by selecting a desired phase setting of cam 124 for discharge at a certain pressure region on the wave curve.

Some advantages arise from employing auxiliary air-intake pipes 120 or 125 of selected length with reference to the sonic wave pattern. By appropriate control of length, a column of the auxiliary air within such pipe may be caused to resonate with such phasing as to assist pumping because of sonic energy transmitted rearwardly through the associated valve during the time that it is opening, open, or closing. A relationship aiding in the operation of the valve is one in which such a pipe has a length corresponding to one half wave length of the sound waves in the contained column of air; although lengths substantially corresponding to a multiple of ⅔ sphere radius can be employed, particularly when the sphere is resonated at its fundamental. One desirable resulting phase response is that which provides a positive pulse in the pipe adjacent the intake valve at opening time.

Figure 6 also discloses an automatic control for the system, operating in response to change in mean pressure in the sonic cavity 51. For this purpose, a control pipe 135 extends into the sphere and terminates in a restricted orifice 136, preferably near a zone of low pressure variation. The control pipe 135 may also contain another restricted orifice 137, these orifices tending to smooth out any remaining pressure undulations so as to supply a constant pressure, substantially corresponding to the mean pressure in the sonic cavity 51, to a bellows 140 working against a spring 141. The bellows is connected by cable or rod 142 to the throttle arm 78. The arrangement is such that the throttle is moved toward open position when the mean pressure falls. On the other hand, when the mean pressure increases, the throttle is closed, ultimately to an idling or stop position. Such a control system is shown with particular reference to the control of combustible mixtures formed by the carburetor 80. If fuel injection is utilized, the bellows 140 can be also connected to any of the well known means for varying the amount of fuel injected with each stroke of the pump 92.

Figure 9 shows an alternative construction of a combustion chamber which can be employed in the embodiments of Figures 1 and 6 to produce sound waves having a more nearly spherical wave front. Here, the orificed body is indicated by the numeral 144 and is, in itself, a small hollow sphere providing a combustion chamber 145 therein. Into this combustion chamber, the combustion-supporting air or the fuel-air mixture is introduced tangentially through the pipe 65, previously described, and no other perforations are provided in the body 144 in the peripheral zone to which this pipe is tangential. Other portions of the body 144 are orificed by employment of short pipes 146 mounted in the walls thereof and extending a slight distance within the combustion chamber for purposes similar to the flanges 57 previously mentioned regarding Figure 7.

Figure 10 shows an embodiment of the invention employing a cylindrically-shaped resonator. Here, a cylindrical wall 150 defines a cylindrical sonic cavity 151 closed at its ends by heads 152. The cylindrical wall 150 provides a curved surface 153 tending to concentrate reflected sound waves toward a focal zone which, in this instance, extends along the central axis of the cylindrical wall 150. A sound wave generator 155 is disposed in the sonic cavity 151, preferably adjacent the focal zone. This generator is shown as including an orificed body 156 in the form of a small cylinder closed by the heads 152 and coaxial with the focal zone.

Increments of fuel are burned intermittently in a combustion chamber 157 within the body 156, this being accomplished, for example, by delivering combustion-supporting air to the combustion chamber 157 through a plurality of intake pipes 160 spaced longitudinally along the axis of the cylinder 150, each pipe being equipped with a valve 161, the opening of which is timed by a corresponding cam 162 on a shaft 163 turned by a suitably governed gasoline engine 164. The governing system may include a governor 165 having an adjustable arm 166 linked to a throttle arm 167 operating a butterfly valve in the induction passage between a carburetor 168 and the cylinder of the engine 164. The shaft 163 is geared to a magneto 170 simultaneously energizing spark plugs 175 disposed along the combustion chamber 157, this energization of the spark plugs being timed with respect to the sound waves. The shaft 163 also drives a fuel-injection pump 180 feeding, by divided lines, a plurality of injection nozzles 181, also spaced along the combustion chamber 157. These nozzles, the spark plugs 175, and the valves 161 are disposed in such relative position that a combustible fuel-air mixture is ignited substantially simultaneously at spaced longitudinal positions of the combustion chamber 157. The products of combustion escape through a multitude of nippled orifices 185, these orifices preferably increasing in size in a direction away from the closest nozzle 181 so that the products of combustion are distributed substantially uniformly into that portion of the sonic cavity 151 around the orificed body 156. Such combustion products establish sound waves in the sonic cavity 151 having substantially cylindrical wave fronts, as compared with the substantially spherical wave fronts in the previously-described embodiments. Such sound waves move outward to, and are reflected by, the cylindrical surface 153, being thereby returned toward, and concentrated in, the focal zone.

A stream of auxiliary air can be supplied through one or more pipes 190, controlled by spring-loaded or cam-driven valves 191, all in the manner previously described. Compressed gases comprising products of combustion can be withdrawn through one or more pipes 192 under control of a valve 193.

Such a system can function on any of the operation cycles previously described. There will be a zone of large or maximum pressure variation in the focal zone. This zone is suggested by the letter P of Figure 10 and a similar zone of pressure variation is indicated at P' near the reflecting surface 153. If the distance between the focal zone and the reflecting surface is approximately ⅝ of a wave length, a single cylindrical zone of large or substantial velocity variation will be established between zones P and P' as indicated by V, and the pipe 192 may desirably discharge compressed gases either from a pressure node region in or near the cylindrical zone V or from a low energy density region near the cylinder wall. It will be understood that the embodiment of Figure 10 is diagrammatically shown and that the control and auxiliary features suggested in Figure 6 may be applied thereto. Also, simplified combustion control apparatus shown for Figure 1 may also be used.

Figure 11 shows a sonic cavity in the form of a spherical sector and comprising a cone 200 having its large end closed by a spherical reflecting surface 201 braced as at 201a and 201b for rigid reflection and having in its small end a transversely positioned grill 202, comprised of a number of horizontal baffles 204 which have an airfoil cross section as seen in Figure 11. A number of fuel nozzles 205 are disposed in rows between baffles 204 with their openings directed into the conical cavity.

Figure 12 shows a sectional view of grill 202 along the line 12—12 as viewed normally from the cavity. The fuel nozzles 205 are supplied with fuel from fuel pump 206 by pipes 207 whenever the air pressure near the fuel nozzle openings falls below a certain delivery pressure. A second grill 203, parallel to the first grill 202 and located directly behind it away from the conical cavity, contains a large number of small rectangular openings 208 as seen in Figure 13. Small metal flaps 209 are attached at the upper and lower edges of each rectangular opening so as to project from the grill in the direction of the conical cavity. The flaps 209 are slightly bowed toward one another so that when unstressed their outer edges touch along a line and they form a pointed arch as shown in the fragmentary perspective view Figure 14. Grill 203 is cast with a small arch-shaped projection 210 from the vertical sides of each rectangular opening. These projections provide seats for the lateral edges of the flaps 209 and also help to form a closure of the rectangular opening when the flaps are in their arched position.

If the pressure within the small end of the conical cavity falls below that on the outer side of the grill 203 the flaps 209 are resiliently bent away from arch-shaped projections 210 and open like double doors (see Fig. 15) to permit the passage of fluid. Reverse flow, however, is effectively checked when the valves assume a closed position. Valves of this type are particularly effective for operation at high frequencies.

When a standing wave exists within the cavity, the small end constitutes a focal zone 212 in which large pressure variations occur. When a rarefaction period begins the flaps 209 open and admit air, which flows between baffles 204 and vaporizes fuel from nozzles 205. If desired, the pressure of the inflowing air may be increased above atmospheric pressure by means of a pump or blower 213.

The occurrence of a rarefaction in focal zone 212 sufficiently lowers pressure in the region of fuel nozzles 205 to permit the discharge of fuel under pressure supplied by fuel pump 206. The entrance of the fuel charge through nozzles 205 is preferably delayed until after the first air introduced has cleared the focal zone 212 of combustion gases remaining from burning the immediately previous charge.

As the rarefaction period draws to a close and the compression half of the cycle begins, the rising pressure closes the flaps 209 and stops or reduces the flow of fuel. The fuel and air charge may be ignited either by spark plug 215 excited by magneto 216, or by flames lingering from the previous combustion or by any other suitable method providing the explosion is timed to reinforce the compression peak of the standing wave. Following the compression part of the cycle, a new rarefaction ensues and the cycle is repeated if the device is operating as a two cycle engine; a scavenging cycle may be allowed to intervene if four cycle operation is desired. To prevent the occurrence of certain harmonics (such as all those above the fundamental) one or more "spoiler" tubes 219 may be used which are open into the cavity while closed at the other end, and of length equal to λ/4 for the frequency being cancelled to cause reflection 180° out of phase. Also these "spoiler" tubes preferably open into a pressure antinode for the undesirable harmonic. The compressed air and combustion products may be withdrawn through a discharge pipe 218 preferably λ/4 long and preferably located toward the large end of the cone in order that gas having a minimum of alternating energy density may be discharged. Also similar pipe 218a in single, multiple, or in the form of an extended slot-passage, may be directed so as to give a radiation pressure propulsion component with the open ends at a velocity antinode.

The solid angle subtended at the apex of the cone may be of any value provided it is not so small that the effect of divergence from the source is negligible and the apparatus functions merely as a closed pipe.

Various auxiliary equipment, such as previously described, may extend into the sonic cavity and the system may be designed for positive mechanical control of fuel and air introduction, ignition, etc., in accordance with the previously described systems.

As seen in Figures 4 and 8, one preferred method of operating resonating cavities, such as the divergent wave cavities disclosed in this specification, provides for introducing a charge of fuel and air into the cavity during each rarefaction period. As shown in the embodiments above described, this may be accomplished either by mechanically opening the intake valves during the rarefaction period or by using spring-loaded check valves in a system in which the fuel and air are supplied at a pressure which remains substantially constant at the check valve which opens only during rarefaction periods. It was assumed that no appreciable sound waves were generated within the intake pipe itself, or that if they were generated, they were so attenuated within the apparatus as to have little effect on the system. It is possible, however, to design a system in which the sound waves generated within the cavity are transmitted into the intake pipe and are reflected therein in such a manner as to produce large variations in pressure at the point of supply to the cavity. These alternations occur with the same frequency as the alternations in pressure within the cavity but are not necessarily in phase with them. These reflected pressure pulses may be used to introduce an increment of fuel or air, or fuel and air mixture, at any time relative to the pressure cycle in the combustion zone within the cavity.

The discussion which follows will be confined to the case in which a gaseous mixture of fuel and air is used but it is to be understood that the same methods may be applied to the introduction of fuel and air separately or of one of them alone. This method is particularly useful in using certain hydrocarbon fuels such as butane, which may be efficiently used when mixed with air before being introduced into the combustion region. Combustible mixtures of air and fuel tend to pre-ignite if brought into the cavity during the rarefaction period. This method provides a means for introducing the bulk of the charge immediately before peak compression.

The pipe used for introducing the fuel into the cavity will be referred to as the intake pipe; its inner end, at which discharge into the cavity occurs, will be referred to as the inner end; its opposite end, whether open or closed, will be referred to as the outer end.

In any of the embodiments herein described, cavity-pipe interconnection such as the intermittent introduction of fuel and air into the cavity, is certain to generate a series of powerful sound wave pulses within the intake pipe. These pulses travel from their point of generation, i. e., the inner end, backwards along the pipe until it is reflected either by encountering a rigid wall surface or by sudden expansion into an enlarged chamber. If the fuel and air supply apparatus has within it a number of small obstructions and apertures at various distances from the opening into the cavity a number of small amplitude waves will be reflected and each will be attenuated so that no substantial pressure variations result therefrom at the inner end. If, however, the intake pipe is smooth and generally free from obstructions, and multiple openings between the cavity intake point and the first region of discontinuity, and if this discontinuity is such as to reflect almost entirely any wave impinging upon it then such reflected waves will return to the inner end to produce large pressure variations thereat. These pressure variations would occur exactly in phase with the pressure variations within the cavity at the pressure antinode region adjacent the pipe's inner end if the length of the intake pipe from its inner end to the discontinuity is precisely a half wave length, or multiple thereof, of the cavity's resonant frequency. If, however, this distance is other than a half wave length, or multiple thereof, the returned pressure pulse will be out of phase with the pressure variations in the cavity's combustion zone. The pipe length may be made adjustable and may be selected so that the reflected pressure pulse occur in such a manner as to force an increment of fuel into the cavity at any desired time with reference to pressure variations in the combustion region. As previously pointed out, best operation with fast burning fuel is achieved if fuel is introduced after the occurrence of rarefaction and just previous to peak compression.

Figure 16 is a diagrammatic illustration of an apparatus for utilizing pressure pulses within the fuel intake pipe to introduce fuel at any desired time. Fuel intake pipe **280** is mounted in the wall of a combustion chamber **10''** (indicated only fragmentarily) and opens at its inner end **281** into a combustion region indicated by the dotted line **13''** which is also a pressure antinode region of cavity **10''**. The outer end **282** of fuel intake pipe **280** is internally threaded to receive the externally threaded section of pipe **283**. The opposite end of pipe section **283** projects into a tank **284** through a closely fitting opening **285**. Tank **284** is supplied with fuel and air mixture entering through pipe **286**. When pipe section **283** is rotated in its threaded connection, its longitudinal movement makes it possible to adjust the distance from the combustion region **13''** within the cavity **10''** to the opening of pipe section **283** into tank **284**. Thus in one mode of operation for example, this distance may be made slightly less than $\frac{3}{8}$ wave length in which case a rarefaction at **13''** will cause an insurge of fuel and air mixture into region **13''** almost ¾ of a period later at a time shortly before peak compression. It will be noted that no valve is necessary for the intermittent introduction of fuel in this device.

In Figure 17 is shown a fuel intake pipe which uses a rigid wall surface in its outer end in order to reflect sound waves. A fuel intake pipe **290** is mounted in the wall of a resonating chamber **10'''**, and projects inwardly so as to open into the resonating cavity at a combustion region indicated by the dashed line **13'''** which is also a pressure antinode region of chamber **10'''**. Fuel and air are supplied to fuel intake pipe **290** through a side branch **291**. The outer end of fuel intake pipe **290** is internally threaded to receive an externally threaded plug **292** which may be threaded in or out of the pipe **290** in order to adjust the length of fuel intake pipe **290** from the opening **294** into combustion zone **13'''** to the reflecting surface **293** on the inner side of plug **292**. If this length is made slightly less than a half wave length of the cavity's resonant frequency then a pressure pulse **13'''** will travel down pipe **290**, be reflected at surface **293** and return up pipe **290** toward combustion region **13'''**, in slightly less than one period so as to introduce an increment of fuel slightly before the compression peak.

It is frequently desirable to prevent backfire into the fuel intake pipe by means of a screen transversely positioned in the pipe near the inner end. Such a screen is indicated by numeral **287** in Figure 16 and by numeral **295** in Figure 17.

In the graph of Figure 18 the solid line **300** indicates the pressure cycle within combustion zone **13'''** and the dashed line **301** indicates the occurrence of pressure pulses just within the inner end of the fuel intake pipe and resulting from the reflection of previous pulses from the end of said pipe. The induction of fuel from the fuel intake pipe into the combustion zone of the resonating cavity at the fuel intake opening is aided when the pressure just within the inner end occurs before the pressure within the cavity a slight distance from the fuel pipe opening. Although the pipe length may be adjusted to cause any desired phasing, including in-phase resonance, in this case the length has been selected so that a reflected pressure pulse **302** appears at the fuel intake opening slightly before the occurrence of peak compression **303** in cavity **10'''**. The maximum intake period is indicated in the graph by the time abscissae **304** to **305**. Although the embodiments of Fig. 16 and Fig. 17 do not employ intake valves, it is to be understood that such valves whether mechanically driven or pressure responsive may under proper circumstances be used in connection with such a system. Any of the embodiments described in previous figures may likewise be converted to a system of this type. For example, the embodiment of Fig. 6 may operate in the manner of Fig. 16 if the turbine casing is sufficiently enlarged, or if a sufficiently large chamber is interposed between the turbine and the air intake pipe **65**, or if air from the turbine enters the air intake pipe **65** from a side branch, the end of the air intake pipe being closed by a solid wall.

In all embodiments of the invention, there will be a remarkably complete combustion of the fuel employed for generating the sound waves. The described sonic standing wave established in the chamber creates very large variations in pressure adjacent the focal zone, and these appear to aid combustion as well as to give large expansion ratios conducive to high thermal efficiency, not to mention the function of these pressure variations in controlling the intake of combustion-supporting or auxiliary air. In addition, the invention for the first time provides a means for creating very large pressure variations in a container without applying equal peak pressures to the inner walls thereof. Still further, the invention is capable of high outputs and excellent thermal efficiencies. One consideration in obtaining high output from the sonic systems herein disclosed is the maintenance of a relatively high mean pressure in the sonic chamber. This mean pressure is controlled largely by the discharge of gases from the sonic cavity and by the automatic throttle control of Figure 6. If this discharge of gases is against a throttling valve or other means for maintaining a back pressure on the system, it is very easy to obtain the mean pressures of two atmospheres or above desired for high output systems.

This invention is not limited to the use of oxygen-free fuels, but comprehends the possibility of injecting explosive type reagents to generate the wave pattern.

In the present specification, the term "sound waves" is used with reference to elastic waves transmitted through a deformable medium, typically gases in the present instance, the waves travelling therethrough at the speed of sound in this medium, the frequency being usually, though not necessarily, within the frequency range of audibility for the human ear. Similarly, terms such as "sonic," acoustic," and similar terms are herein used without limitation as to humanly audible frequencies. Actual test operation has proven that many practical sizes and modes of operation of the apparatus results in frequencies recognized as within the human audible range. For human comfort, this invention comprehends the provision of sound deadening covering over the apparatus as well as conduit mufflers such as standard engine exhaust mufflers connected with the various intake and discharge pipes.

Certain of the broader subject matter of the instant application is disclosed and claimed in my copending applications, Serial Nos. 439,926 and 589,754, both now abandoned, which disclose and claim certain of the broader aspects of a thermally excited resonator without limitation to the employment of diverging waves only.

Various changes and modifications including combining any one or more embodiments of this invention into series multi-stage or parallel multiple combinations can be made without departing from the spirit of the invention as defined in the appended claims.

I claim as my invention:

1. In combination: walls defining a closed cavity and a substantially curved sound-wave-reflecting surface shaped to reflect sound waves originating within a predetermined zone spaced from said reflecting surface, toward a focal zone of said reflecting surface, said cavity being adapted to contain a fluid; a thermally excited sound-wave generator located within said predetermined zone for establishing sound waves in said fluid, said generator being operable at a frequency high enough to produce sound waves whose quarter wave length is less than the distance between focal zone and reflecting surface, and which is such as to establish standing wave resonance, with sound waves launched toward said reflecting surface reflected and concentrated in said focal zone to produce a pressure anti-node at said focal zone and a velocity anti-node between said focal zone and said reflecting surface; means for intermittently introducing fluid to said pressure anti-node region of said cavity in response to decreasing pressure of said sound waves; and discharge means leading from the velocity anti-node region of said cavity.

2. In combination: walls defining a closed cavity and a substantially curved sound-wave-reflecting surface disposed geometrically about a focal zone to reflect toward said zone sound waves which originate therein, said cavity being adapted to contain a fluid; a thermally excited sound-wave generator located within said closed cavity at a position adjacent said focal zone for generating said waves, said generator being operable at a frequency to produce standing wave resonance in the cavity, with a wave train moving from said generator through said fluid to said reflecting surface to be reflected to and concentrated in said focal zone, all in a manner to produce a pressure anti-node region at said surface, another pressure anti-node region at said focal zone, and a velocity anti-node between said pressure anti-nodes; means for intermittently introducing fluid to the pressure anti-node region at said focal zone of said cavity in response to decreasing pressure of said sound waves; and fluid discharge means leading from said velocity anti-node region of said cavity.

3. A combination as defined in claim 2, in which said reflecting surface is substantially spherical in shape, and wherein the radius of said sphere is greater than a quarter wave length.

4. A combination as defined in claim 2, in which said reflecting surface is substantially cylindrical in shape to provide an elongated focal zone, and in which said sound-wave generator is elongated to occupy a substantial portion of the length of said focal zone.

5. A combination as defined in claim 2 in which said cavity is substantially conical in shape and said reflecting surface comprises the base of said conical cavity and corresponds substantially to the surface of a sphere having its center at the apex of said conical cavity.

6. In combination: walls defining a cavity, said walls including a reflecting surface shaped to reflect sound waves originating adjacent a focal zone of said reflecting surface, said cavity being adapted to contain a fluid; a sound wave generator within said closed cavity at a position adjacent a focal zone of said surface; means for operating said sound wave generator at a frequency which resonates said cavity; means communicating with a pressure antinode region for delivering a fluid to said closed cavity during periods of low pressure; and discharge means for withdrawing fluid from said closed cavity at a position near a zone of substantial velocity variation.

7. In combination: walls defining a cavity bounded at least in part by a reflecting surface shaped to reflect resonant sound waves therein toward a region of relatively high energy density; means for resonating fluid in said cavity; intake means for introducing fluid into said cavity near a region of large pressure variation during rarefaction periods at said region; discharge means leading from a pressure antinode region of relatively low energy density, said discharge means being rearwardly directed to provide a propulsive jet and having a length sufficient to produce a velocity antinode at the outer opening to produce alternate jetting and sucking.

8. In combination: walls defining a cavity bounded at least in part by a reflecting surface shaped to reflect sound waves toward a region of relatively high energy density; means for introducing fluid into said cavity; means for resonating fluid in said cavity; and brace means for connecting said reflecting surface walls to other support means, said brace means being sufficiently rigid to substantially reduce any vibration of said reflecting surface; and means for discharging compressed fluid from said cavity.

9. In combination: walls defining a cavity bounded at least in part by a reflecting surface shaped to reflect sound waves originating in a focal region of said reflecting surface toward a region of relatively high energy density; means for resonating fluid in said cavity; intake means for introducing fluid into said cavity near a region of large pressure variation during rarefaction periods at said region, said intake means comprising a plurality of reed valves having a frequency not less than the resonant frequency of the cavity; and discharge means from said cavity.

10. A combination as in claim 9 in which said

reed valves are disposed over a surface in the shape of the wave front to be generated.

11. In combination: a substantially curved sound-wave-reflecting surface having a focal zone and so shaped that sound waves emanating from said focal zone will move outward to said surface to be reflected toward and concentrated in said focal zone; means for burning increments of fuel in said focal zone at such frequency as to establish a condition of acoustic resonance in the space between said focal zone and said reflecting surface with a first pressure antinode region adjacent said reflecting surface and second pressure antinode region adjacent said focal zone, there being at least one velocity antinode region between said reflecting surface and said focal zone; means for supplying fuel to said fuel burning means; and means for withdrawing products of combustion from said velocity antinode region.

12. A combination as defined in claim 11, in which said substantially curved sound-wave-reflecting surface bounds at least a portion of a closed cavity, and including means for restricting the discharge of said products of combustion from said velocity antinode region to maintain a main pressure in said closed cavity.

13. In combination: a substantially curved sound-wave-reflecting surface having a focal zone; walls defining a combustion chamber adjacent said focal zone; a movable member adjacent said combustion chamber; means for introducing fuel to said combustion chamber; means for burning increments of fuel in said combustion chamber to create pressure pulses therein acting upon one side of said movable member to move same to aid in establishing a sound wave moving to said reflecting surface; and fluid discharge means leading from said cavity.

14. In combination: an internal-combustion resonator including a sonic cavity bounded at least in part by a substantially curved reflecting surface; means for introducing fuel to said cavity; means for intermittently burning increments of said fuel in said cavity to establish therein substantially a condition of acoustic resonance to create pressure undulations in a focal zone of said reflecting surface; a shaft; means for turning said shaft at a speed proportional to the pressure undulations in said cavity, said means for intermittently burning said increments of fuel including ignition means timed by the rotation of said shaft for igniting said increments of fuel; and fluid discharge means leading from said cavity.

15. In combination: walls defining a sonic cavity bounded at least in part by a reflecting surface disposed about a focal region; a fuel-introducing means for intermittently supplying increments of fuel to said cavity; a turbine receiving products of combustion from said cavity and operatively connected to said fuel-introducing means; means for initiating combustion of said fuel increments in said sonic cavity to form said products of combustion; and means for controlling the speed of said turbine to control the rate of fuel introduction in a manner to establish substantially a condition of acoustic resonance in said sonic cavity.

16. In combination: walls defining a sonic cavity bounded at least in part by a reflecting surface; means for introducing into and burning in said cavity increments of fuel at such frequency as to establish a condition of acoustic resonance in said chamber, said reflecting surface being shaped to reflect sound waves toward and to establish large pressure variations in a zone adjacent the point of fuel introduction; means for timing the introduction and burning of said fuel increments to occur at a frequency lower than but proportional to the resonant frequency of the sound waves in said sonic cavity; and fluid discharge means leading from said cavity.

17. In combination: walls defining a sonic cavity bounded at least in part by a reflecting surface, said reflecting surface being shaped to reflect sound waves originating at a combustion zone within said sonic cavity back towards said combustion zone; means including fuel feeding and burning means for initiating a first explosion in said combustion zone, said first explosion tending to establish a train of sound waves moving to and fro in said sonic cavity between said combustion zone and said reflecting surface to create pressure pulses in said combustion zone at a frequency proportional to the natural resonant frequency of said sonic cavity, the train of sound waves initiated by said first explosion tending gradually to die down; control means operating in step with said sound waves; said fuel feeding and burning means being operable by said control means to initiate succeeding explosions in said combustion zone in timed reinforcing relationship with pressure pulses in said zone because of prior explosions but with the occurrence of at least one un-reinforced pressure pulse in said combustion zone between any two explosions; and fluid discharge means leading from said cavity.

18. A combination as defined in claim 17, including means for admitting a gas to said sonic cavity during rarefaction periods.

19. A combination as defined in claim 17, in which said explosions are created by burning increments of fuel in said combustion zone, and including means operated by said control means for delivering an increment of fuel to said combustion zone during the rarefaction periods preceding each explosion.

20. A method of intermittently burning increments of fuel in a sonic cavity having a reflecting surface shaped to reflect sound waves toward a focal zone of high energy density, which method includes the steps of: introducing said increments of fuel into said focal zone and burning same therein at a resonant frequency of said cavity such as to generate sound waves whose quarter wave length is less than the distance between reflecting surface and focal zone and thereby establish at said focal zone a pressure anti-node characterized by large variations in pressure; and withdrawing products of combustion from said sonic cavity at a position spaced from said focal zone and at which position energy density is substantially less than in said focal zone.

21. In combination: walls defining a chamber the interior of which is substantially spherical in shape; combustion means located near the center thereof; means for supplying said combustion means with a combustible mixture; means for causing periods of maximum combustion to occur at said combustion means with a frequency to resonate fluids contained within said hollow spherical chamber, said frequency being such as to produce sound waves of wave length substantially 1.4 times the radius of said spherical chamber, so as to establish a standing wave in the fluid in the spherical chamber characterized by creation of pressure anti-nodes in the center region of the spherical chamber and adjacent the exterior wall thereof and an intervening velocity

anti-node; means for introducing fluid to a pressure anti-node region of said chamber; and fluid discharge means leading from said velocity anti-node region of said spherical chamber.

22. In combination: walls defining an enclosed chamber, the interior of which is substantially conical in shape; combustion means located near the apex thereof; means for supplying said combustion means with a combustible mixture; means for causing periods of maximum combustion to occur at said combustion means with a frequency to resonate fluids contained within said hollow sphere, and fluid discharge means leading from said cavity.

23. In a resonant combustion chamber pump of the type described herein, a means for intermittently introducing a combustible mixture into said resonating combustion chamber which includes: a mixture intake pipe which is adjustable in length, one end opening into said resonating combustion chamber and the other end being

closed; and means for introducing combustible mixture into said intake pipe.

24. In a resonant combustion chamber pump of the type described herein, a means for intermittently introducing a combustible mixture into said resonating combustion chamber which includes: a mixture intake pipe which is adjustable in length, one end opening into said resonating combustion chamber and the other end opening suddenly into a region of substantially larger cross-section; and means for introducing combustible mixture into said intake pipe.

ALBERT G. BODINE, JR.

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Number	Country	Date
176,838	Great Britain ------	Mar. 16, 1922

Oct. 1, 1929. T. BELLOCQ 1,730,336

APPARATUS FOR THE EXTRACTION OF LIQUIDS

Filed Feb. 6, 1924

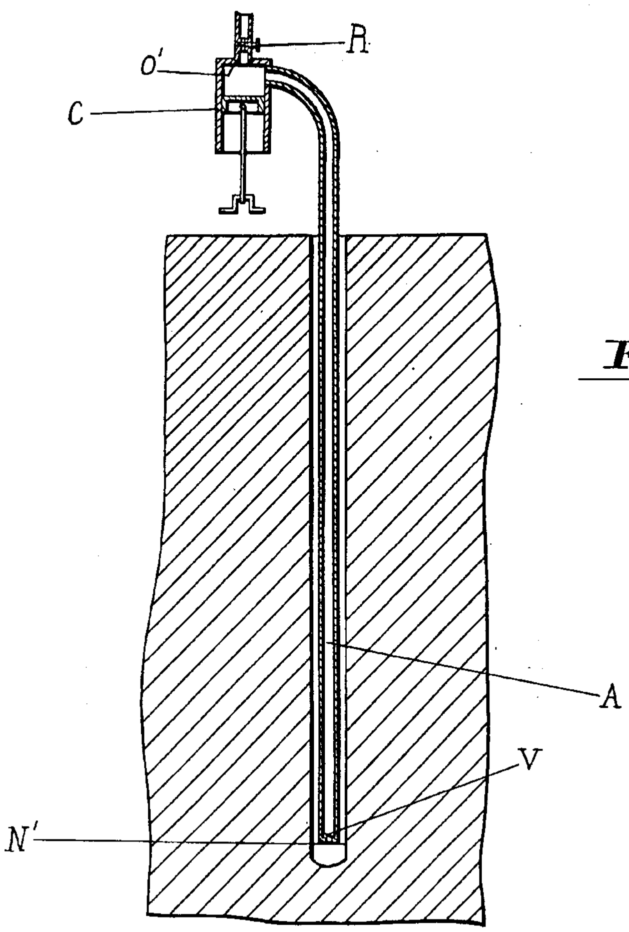

Fig.1.

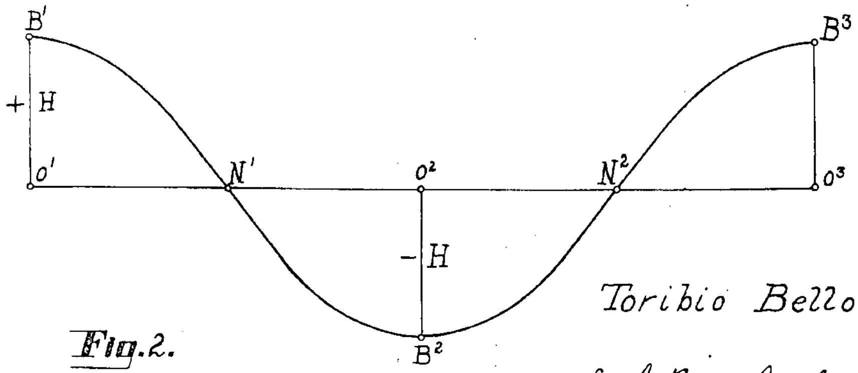

Fig.2.

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By Emil Bönnelycke

Attorney

Patented Oct. 1, 1929 **1,730,336**

UNITED STATES PATENT OFFICE

TORIBIO BELLOCQ, OF BUENOS AIRES, ARGENTINA

APPARATUS FOR THE EXTRACTION OF LIQUIDS

Application filed February 6, 1924, Serial No. 691,068, and in Argentina December 3, 1923.

My invention relates to moving or transporting liquids and especially to extracting or pumping liquids from deep wells or other source of supply in a new manner; its main purpose being directed towards a pumping scheme of very simple and economical construction by which the mechanism at the bottom of the well is reduced to a minimum and the extraction of fluid effected from the ground level substantially irrespective of the depth of the well, and thus when used for water it is not limited to the 25 or 30 foot limit of the usual suction pump for water, and similarly the limits usual with pumps for other liquids do not persist. In order that my present invention may be clearly understood and easily carried into practice, a preferred embodiment thereof has been shown in the appended diagrammatic drawings wherein Figure 1 is a vertical section of a well and apparatus and Fig. 2 is a diagram of the curve of the pressure wave to which reference will be made hereinafter.

In the embodiment shown the device consists of a pipe A through which the liquid is withdrawn. The pipe reaches to the bottom of the well, or to the liquid level therein, and is of a suitable length as will be explained. At the bottom of the pipe is a check or retention valve V. The upper end of the pipe A is connected at or about the ground level with a pump or compressor C preferably making rapid reciprocations with a short stroke. The compressor is a valveless pump and is provided with an outlet controlled by an adjustable cock or the like R, through which the pumped liquid may pass from the system. The cock may be opened at the beginning of the operation but may be adjusted to a desirable point during operation.

Before starting the operation of the compressor care must be taken to completely fill the pipe A and the cylinder of the compressor with liquid, carefully excluding substantially all free air and being sure that the piston of the compressor is in the outermost position.

It is well known that energy may be conducted through a column of liquid by setting up a wave motion therein. The waves so set up are comparable to sound waves or waves of electric energy. In order to set up such waves it is necessary to cause alternate areas of high and low pressure in the medium and this may be effected by any suitable known means. I find it convenient to use for this purpose a reciprocating piston. The piston of the compressor C when rapidly reciprocated sets up waves in the liquid in the pipe A by alternately compressing the liquid in the pipe and releasing the pressure especially when the piston has a short stroke. These waves have a length L equal to $\frac{v}{n}$ in which n is the number of strokes per second of the piston of the compressor C and v is the speed of the wave per second. When operating in water the speed of the wave may be substantially the speed of sound in the water under the operating conditions.

As may be seen from the diagram in Fig. 2 the wave form includes bulges B^1, B^2, B^3 . . . which are associated with corresponding nodes N^1, N^2, N^3 . . . . When the closed end of the pipe is at a distance from the reciprocating piston equal to any number of half waves, that is at O^1, O^2, O^3 . . . or when the pipe is cut off at a distance equal to any odd number of quarter waves, that is at N^1, N^2, N^3 . . . , a stationary wave may be set up in the pipe. For the purpose of the present invention it is preferred to arrange the valve V at the end of the pipe and adjust the apparatus in such a way that the valve will be at an odd quarter wave length such as N^1, or N^2 etc., but the purpose can be accomplished, possibly less efficiently, by other adjustments.

The maximum variation of pressure accompanied by no variation in flow of the liquid occurs at the points O^1, O^2, O^3 . . . whereas at the nodes N^1, N^2, N^3 . . . the variation in pressure is zero with the maximum variation in flow of the liquid. By the preferred arrangement of placing the valve V at the point N^1 or N^2 etc. the liquid may enter continuously. The valve V tends to remain always open but it may open and close from time to time during the operation. On the other hand the liquid will flow more or less intermittently from the cock R when it is adjusted to the proper opening. The required pressure in the apparatus may be regulated by choosing an appropriate diameter of pipe and by employing a piston of the proper cross-section, stroke and speed. If required or thought desirable or necessary there may be installed a liquid filled bottle or chamber to act as a capacity or condenser in the manner well known in connection with the transmission of energy by means of waves traveling in liquids.

I have found that one specific appropriate apparatus which will illustrate in a concrete way the present invention may consist of a fluid pipe having an internal diameter of one inch. Such a pipe placed in a well having water at a depth of 20 meters may have arranged at its lower end a valve opening 30 millimeters in diameter seated in which is a ball of approximately 38 millimeters in diameter and held in place by the usual cage which may be adjusted so that the ball can rise from its seat about 20 millimeters. On the surface of the earth the pipe may be led into the cylinder of a compressor of 50 millimeter diameter in which is a piston having a stroke of about 36 millimeters. The piston may be reciprocated at a speed of about 360 revolutions per minute by an electric motor. Leading out from the cylinder of the compressor may be a pipe of one half inch internal diameter in which is placed an adjustable cock or valve. The cock will not be closed but is used merely for adjusting the out flow. The piston will be moved to its outermost position and the cylinder and pipe entirely filled with water, the cock being left open. At this time the ball valve will be closed. On starting the electric motor the ball valve will lift from its seat and may remain open throughout the operation when the cock is properly adjusted, but it may vibrate from its seat opening and closing during the operation. Water will begin to flow from the outlet and by an appropriate adjustment of the cock a continuous operation may be effected and an output of 1,000 liters per hour procured.

While the fundamental theory on which the operation is based may be somewhat in doubt, I believe that the rapid reciprocation of the piston working upon the water in the apparatus produces a series of periodic pressure variations with periodic changes of pressure and volume throughout the liquid column due to the elasticity and compressibility of the liquid. The energy waves so set up travel to the valve V at which point they may be reflected and the transmitted energy is sufficient to open the valve V and lift the column of liquid at the same time drawing in liquid from the well itself. In some systems of energy transmission through liquid there is no substantial flow of the main body of liquid for carrying the energy. I have discovered, however, that apparently the transmission of energy may go through a flowing column of liquid and the transmitted energy may be relied upon to keep the liquid in more or less constant flow.

According to the present invention liquids may be extracted from substantially any depth without using any complicated machinery or mechanism in the well and simply by lowering a pipe of the required length and dimensions duly connected to a simple compressor the latter being at or about the ground level. The valve at the lower end of the pipe will be subject to substantially no wear if the valve is substantially always in its open position during the operation of the compressor. On starting this valve is closed and great care must be taken when filling the pipe and cylinder with liquid to see that substantially no free air remains in the system.

The piston may work either with the cylinder horizontal or with the cylinder vertical or inclined. The pipe through which the liquid flows may be either horizontal or vertical or inclined and it may be curved or straight. All of these arrangements may be referred to as pumps and I use the term pumping as including moving, transporting or conveying in vertical or horizontal or inclined direction.

The term liquids is used to include not only simple liquids but also mixtures of liquids and such mixtures of liquids with gases or solids as are capable of being transported through pipes or pumped.

It is obvious that the device has been shown only as a preferred embodiment and that any other may be used for the same purpose. Apparatus for carrying out the invention may be constructed from mechanisms or devices already known and used for other purposes and the invention is not confined to any specific form of valves or compressors or means of operating them.

I claim:

1. An apparatus for pumping liquids comprising a pipe filled with liquid leading to the liquid supply, a check valve in the end of the pipe in the liquid supply, a compressor cylinder at the other end of the pipe, an open outlet from the cylinder, and a piston in the cylinder and means to cause the piston to alternately compress the liquid in the pipe and release the pressure.

2. An apparatus for pumping liquids comprising a pipe leading to the liquid supply and filled with liquid, a check valve in the pipe within the liquid to be moved, an outlet for the pipe open to the air, and means for alternately compressing the liquid in the pipe and releasing the pressure.

3. Apparatus for pumping liquids comprising a check valve in the liquid supply, a compressor adjacent the point of delivery, a pipe filled with liquid leading from the valve to the compressor, an outlet for the liquid, and means for operating the compressor to

alternately compress the liquid in the pipe and release the pressure to operate the check valve and move the liquid.

4. Apparatus for pumping liquids comprising a check valve in the liquid supply, a compressor at the point of delivery including a cylinder and a piston and an outlet, a pipe filled with liquid extending from the valve to the compressor, and means for reciprocating the piston rapidly through a short stroke whereby the compression of the liquid produced by the piston is varied.

5. The method of pumping liquids comprising alternately compressing a confined body of liquid in a pipe and releasing the pressure and causing the pressure variations to move the liquid toward the compressing means and permit new liquid to enter the pipe from a source of supply.

6. The method of pumping liquid comprising placing in the liquid supply a pipe provided with an outlet near one end and carrying at the other end a check valve in the liquid supply, filling the pipe with liquid, and alternately compressing the liquid in the pipe and releasing the pressure to cause the liquid to flow toward the compressing means near the outlet and to cause new liquid from the supply to enter the pipe.

7. The method of pumping liquids in pipes having outlets comprising repeatedly compressing liquid in the pipe and releasing the pressure, and causing the pressure variations to pass through the liquid so as to cause liquid to enter the pipe and pass through it.

8. Apparatus for pumping liquids comprising means for alternately compressing a body of liquid in a pipe and releasing the pressure, and an outlet for the pipe, whereby the variations in compression cause liquid to enter and flow through the pipe.

9. The method of pumping liquid through a pipe provided with an outlet near one end and a check valve at the other end, comprising placing the check valve in the liquid to be pumped, filling the pipe with liquid and alternately compressing and releasing the liquid in the pipe.

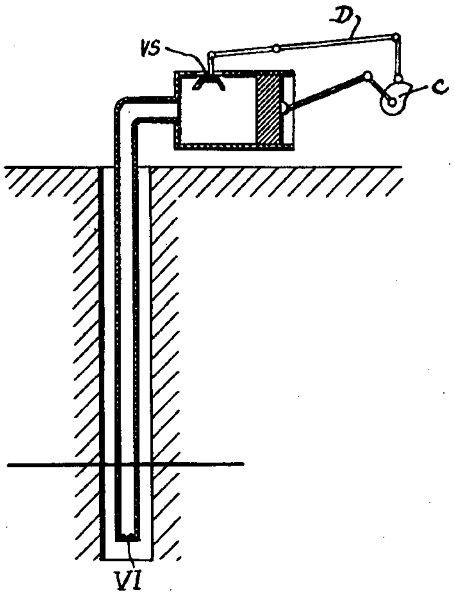
In testimony whereof I affix my signature.

TORIBIO BELLOCQ.

Oct. 1, 1929.　　T. BELLOCQ　　1,730,337

PUMP

Filed Aug. 30, 1928



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ATTORNEY.

Patented Oct. 1, 1929 **1,730,337**

UNITED STATES PATENT OFFICE

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PUMP

Application filed August 30, 1928. Serial No. 303,074.

My invention relates to a new type or scheme of moving or transporting liquids and especially to pumping permitting of the elevation of liquids from substantially any depth without the necessity of situating the machinery at the bottom of the well and includes the employment not only of the energy produced by the waves set up by the rapid reciprocation of a piston but also the extra waves or impulses produced by the opening and closure of a valve in the system. Variations of pressure in the body of liquid in the pump may be employed to effect the pumping and it is more efficient and a larger output is produced if the variation is produced in a plurality of series.

Figure 1 of the accompanying drawing illustrates a vertical section through apparatus incorporating the invention.

The apparatus for carrying out the invention consists of a pipe reaching to the bottom of the well or within the water therein and provided at its lower end with a check valve VI. At or about the ground level the pipe enters the cylinder of a pump or compressor provided with a piston. The cylinder has a single valve VS. In beginning the operation the piston is placed at its outermost position and the pipe and cylinder are completely filled with liquid, care being taken to exclude substantially all free air. The valve VI prevents the liquid running out of the pipe and the valve VS is closed. The piston is then pushed forward by a suitable power and its reciprocation started. After the piston has moved a proper part of its stroke the valve VS is opened and liquid issues therefrom. The valve VI also opens and liquid enters the bottom of the pipe from the well and the continued operation of the device lifts liquid from the well bottom and delivers it through the valve VS which is closed at the proper part of each stroke of the piston at which time the outflow of liquid will be temporarily stopped.

I have thus described the physical operation of the apparatus, while the underlying theory of its operation and the theoretical application of the physical forces involved may not be entirely clear, there will now be set forth a full and clear statement of what is believed to be the explanation of the operation and how the physical forces interact with the apparatus to produce the desired effect.

The apparatus employed involves three parts, namely (1) the apparatus for producing the impulses or waves of force and for producing the opening and closing extra waves or impulses (2) the transmission line or pipe (3) a valve in the lower end of the pipe which may produce in its turn opening and closing extra waves or impulses.

1. The first portion of the apparatus may consist of a compressor of any approved design having a single valve VS diagrammatically illustrated in Fig. 1. The piston of the compressor is rapidly reciprocated preferably through a short stroke so as to produce waves of energy in the liquid column. Initially the piston is moved to its outermost position and the pipe and compressor cylinder filled with liquid. The valve VS is initially closed. The compression of the liquid caused by the inward stroke of the piston initiates a wave movement through the column of liquid. When the movement of the piston has produced the desired pressure, which may be before or at the end of its stroke, the valve VS is suddenly opened. The cooperation and relative timing of the piston and valve may be accomplished in any appropriate way but I prefer to have them mechanically interlocked so that the valve will open at the appropriate moment in the piston stroke. When this valve opens it suddenly, by the variation of pressure, transforms the piston wave into an opening extra impulse or wave in the liquid column. Advantage may be taken also of the extra wave caused by the closure of the valve but the closure extra waves are of lesser effect than the opening extra waves. Valve VS being closed when the piston enters the cylinder a certain pressure is produced to set up a wave in the liquid column. When the valve VS opens the opening sets up an extra or supplemental wave impulse which like the piston wave will travel through the whole length of the liquid in the pipe to the lower valve VI. This extra or auxiliary wave

or impulse may be similar also to that produced in water rams. The water ram, however, takes advantage of the closing extra wave rather than the extra wave produced by opening. In my system this extra wave when working with the valve VS open to the air is probably one of much volume and slight pressure. This wave and extra wave cause the liquid to proceed from the well by valve VI and carry it through the pipe and and drive it through valve VS which is open. It is necessary to calculate the necessary pressure and velocity of the wave for the length and diameter of the pipe employed as a guide in altering or adjusting, in each case, the diameter, stroke and frequency of the piston. As is well known in systems transmitting energy by wave through columns or pipes of liquid the pressure produced in a tube or pipe of a given diameter may be varied or controlled by providing in communication with the pipe a further container filled with the same liquid which is being pumped to act as a capacity or condenser. In order to set up such waves it is necessary to cause alternate areas of high and low pressure in the medium and this may be effected by any siutable known means. I find it convenient to use for this purpose a reciprocating piston and valve as indicated.

2. The second portion of the apparatus consists merely of a suitable pipe which brings up the liquid and, through the flowing stream, transmits the energy or waves from the piston and the upper valve VS to the lower valve VI it being necessary to calculate the section of the pipe in accordance with the pressure and the volume of the liquid to be obtained or extracted from the well. Since energy-bearing wave-currents in liquid have a certain velocity, the frequency of the piston must be calculated in order that the wave produced should conform with the length of the pipe, it being possible to work likewise with the harmonics of the wave and also to work when the pipe length corresponds with odd or even fourths of such wave length. This is preferred but the purpose can be accomplished, possibly with less efficiency, with other adjustments.

3. The third portion of the apparatus consists of the lower valve VI which may serve three purposes, (*a*) it serves as a non-return valve so as to permit the pipe being filled up before starting; (*b*) it causes the liquid to move in one direction only avoiding the return thereof into the well; and (c) it may be that it also serves as a wave or impulse transformer like the upper valve VS producing by its movement an extra wave or impulse. Thus the pipe will draw in a great quantity of liquid under low pressure being the inverse of the piston which produces a wave or impulse of high pressure and little volume.

At first sight it might seem that if the piston is caused to enter the cylinder when the valve VS is closed the apparatus would burst. This is not the case, however, since the capacity for compression and the elasticity of the liquid being pumped, when known, serve to allow the stroke of the piston and the pressure obtained to be calculated and the apparatus to be operated in such a way as not to exceed the limit of resistance of the apparatus before the opening of the valve. It will be remembered that the impulses and extra impulses of energy in the wave form are possible solely because the liquids worked upon are compressible and elastic which permits of their working in a resilient manner.

The operation of the apparatus is as follows. The piston is withdrawn from the cylinder to the maximum extent possible and the cylinder and pipe are filled up substantially completely with liquid taking care that substantially no free air be left inside. The valve VS is closed and the piston is pushed inwardly vigorously causing a compression of the liquid and initiating an energy carrying wave which moves through the column of liquid. As soon as the piston has entered sufficiently to produce the desired pressure the valve VS is suddenly opened. This abrupt opening of the valve VS permits escape of liquid and suddenly changes the pressure thus producing an opening extra wave which will travel through the whole length of the pipe. The waves produced by the piston and by the valve VS both carry energy to the valve VI which will be opened. The system is more economical and more efficient than one which employs for the pumping and lifting of the liquid only the energy transmitted by the waves set up by the piston action. If the opening of the valve VI is abrupt it may transform the extra wave into a wave of small pressure but great volume. Since the valve VI does not permit the liquid to flow downwards an upward flow will occur, that is to say, the energy transmitted by the two waves will not only open the valve VI but will lift the liquid in the pipe and draw from the well liquid past the valve VI and upwards through the pipe from which it will discharge through the open valve VS. Operation then may become substantially continuous the piston being rapidly reciprocated and the valve VS operated in harmony therewith being always closed at the proper part of the piston stroke. It might be best to arrange the shaft moving the piston so that it would also govern the valve VS. Thus the work might be done in a completely automatic manner. Thus a cam C may be mounted on the drive shaft and engage mechanism D to operate the valve VS at the proper predetermined point in the stroke of the piston. Any other suitable means for associating the operation of the piston and the valve may be employed. The time of opening and closing the valve VS and its size may be adjusted to give the maximum output of liquid with the minimum expense of energy to operate the piston. The apparatus expels during each stroke of the piston a volume of liquid several times greater than the volume displaced by the piston. In practice it is possible to expel during each stroke of the piston a volume of liquid about seventeen times as large as the volume displaced by the piston and for this reason the stroke of the piston may be very short or the volume displaced at each stroke of the piston may be small.

I have found that one specific appropriate apparatus which will illustrate in a concrete way the present invention may consist of a fluid pipe having an internal diameter of one inch. Such a pipe placed in a well having water at a depth of 20 meters may have arranged at its lower end a valve opening 30 millimeters in diameter seated in which is a ball of approximately 38 millimeters in diameter held in place by the usual cage which may be adjusted so that the ball can rise from its seat about 20 millimeters. On the surface of the earth the pipe may be led into the cylinder of a compressor of 50 millimeters diameter in which is a piston having a stroke of about 30 millimeters. The piston may be reciprocated at a speed of about 180 revolutions per minute by a suitable electric motor. In the cylinder of the compressor may be a valve having a diameter of about 30 millimeters and adjusted so as to have a movement of from 5 to 7 millimeters. The piston will be moved to its outermost position and the cylinder and pipe entirely filled with water. The ball valve rests by gravity in its closed position and the valve in the cylinder is closed. The electric motor will be started and at the proper time in the stroke of the piston the valve in the cylinder will be opened. The ball valve will open and water will begin to flow from the piston valve with an output of about 4,000 or 5,000 liters per hour.

While the fundamental theory on which the operation is based may be somewhat in doubt, I have endeavored to set out herein what I believe is the true principle but my invention is not to be so limited. In order to set up such waves as are here involved it is necessary to produce variations in compression in the liquid and although a compressor is illustrated for such purpose it will be understood that the invention is not so limited but extends to any suitable or known means for producing such effect.

This system permits of working at any desired depth without being compelled to arrange the machinery within the well, provided the necessary pressure has been well calculated. It is possible that advantageously, the work is done under a considerable pressure in order to use, same as in electricity, the transmission of high voltages over long distances with favorable output. This system also permits of working with a piston of reduced dimensions as, taking advantage of the transforming effect of the wave or impulse, a piston of a given displacement, under a given pressure, would be able to elevate at each stroke a liquid volume several times greater than the same displacement but under a pressure several times lower.

This system would seem to be contrary to the laws of gravity which does not permit of liquid being elevated or drawn up from a depth greater than the length of a column of the same liquid counter-balancing the atmospheric pressure; however, in fact there is employed a force in wave form which travels throughout the length of the pipe and arrives at the lower valve VI. This force operates at this valve in the same manner as if there were a piston at this point.

The present system would seem also to be contrary to the law which does not permit a piston of a given displacement to obtain at each stroke a volume of liquid greater than such displacement; but it will be apparent that a piston of a given displacement, under a given pressure, may obtain at each stroke a liquid volume several times greater than the volume of displacement of such piston, but under a pressure several times less.

The piston may work either with the cylinder horizontal or with the cylinder vertical or inclined. The pipe through which the liquid flows may be either horizontal or vertical or inclined and it may be curved or straight. All of these arrangements may be referred to as pumps and I use the term pumping as including moving, transporting or conveying in vertical or horizontal or inclined direction.

The term liquids is used to include not only simple liquids but also mixtures of liquids and such mixtures of liquids with gases or solids as are capable of being transported through pipes or pumped.

It is obvious that the device has been shown only as a preferred embodiment and that any other may be used for the same purpose. Apparatus for carrying out the invention may be constructed from mechanisms or devices already known and used for other purposes and the invention is not confined to any specific form of valves or compressors or means of operating them.

Having thus fully described and ascertained my said invention, and the manner in which the same is to be performed and carried into practice, I declare that what I claim and desire to secure by Letters Patent is:—

1. Apparatus for pumping liquids comprising a pipe filled with liquid leading to the liquid supply, a check valve in the end

of the pipe in the liquid supply, a compressor cylinder at the other end of the pipe, an outlet for the liquid, a piston in the cylinder, means to cause the piston to repeatedly compress the liquid in the pipe, a valve in the outlet and means for opening the outlet valve at a predetermined point in each stroke of the piston.

2. Apparatus for pumping liquids comprising a pipe leading to the liquid supply and filled with liquid, a check valve in the pipe within the liquid to be moved, an outlet for the liquid, means for compressing the liquid in the pipe, and separate means for suddenly releasing the pressure.

3. Apparatus for pumping liquids comprising a check valve in the liquid supply, a compressor adjacent the point of delivery, a pipe filled with liquid leading from the valve to the compressor, an outlet for the liquid, means for operating the compressor to compress the liquid in the pipe, and separate means to suddenly release the pressure, to operate the check valve and move the liquid.

4. The method of pumping liquids comprising placing in the liquid supply a pipe provided with an outlet near one end controlled by a valve and carrying at the other end a check valve in the liquid supply, filling the pipe with liquid, compressing the liquid in the pipe, and operating the valve controlling the outlet to suddenly release the pressure to cause the liquid to flow toward the compressing means near the outlet, and to cause new liquid from the supply to enter the pipe.

5. The method of pumping liquid through a pipe provided with a controlled outlet near one end and a check valve at the other end, comprising placing the check valve in the liquid to be pumped, filling the pipe with the liquid, and repeatedly compressing the liquid in the pipe and operating the outlet controlling means to suddenly release the compression.

6. Apparatus for pumping liquids comprising a pipe leading to the liquid supply and filled with liquid, a check valve in the pipe within the liquid supply, an outlet for the liquid, and means to set up a contemporaneous plurality of series of compression waves in the liquid in the pipe to operate the check valve and move the liquid.

7. The method of pumping liquids in pipes comprising repeatedly compressing the liquid to cause a series of impulses of energy in the form of compression waves to pass through the liquid and repeatedly suddenly relieving the compression of the liquid to cause another series of impulses of energy in the form of compression waves to pass through the liquid to cause liquid to pass through the system.

8. Apparatus for pumping liquids comprising a pipe leading to the liquid supply and filled with liquid, a check valve in the pipe in the liquid supply, a compressor cylinder at the other end of the pipe, a valve in the cylinder, a piston in the cylinder and means for operating the piston and the valve in the cylinder to set up a plurality of series of compression waves in the liquid in the pipe to operate the check valve and pump the liquid.

9. The method of pumping liquid through a pipe provided with a controlled outlet near one end and a check valve at the other end, comprising placing the check valve in the liquid to be pumped, filling the pipe with the liquid, and applying energy to the liquid in the pipe in a contemporaneous plurality of series of impulses in the form of compression waves at a point toward which flow occurs.

10. Apparatus for pumping liquid comprising means for alternately compressing a body of liquid in a pipe, separate means for alternately releasing the pressure, and an outlet for the pipe, whereby the variations in compression cause the liquid to enter and flow through the pipe.

In testimony whereof I affix my signature.

TORIBIO BELLOCQ.

Jan. 2, 1934. T. BELLOCQ **1,941,593**

PUMPING

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Fig.3.

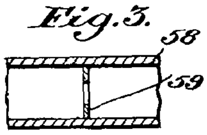


Fig.4.

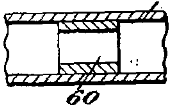


Fig.5.

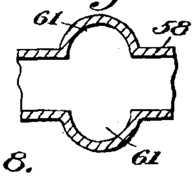


Fig.6.

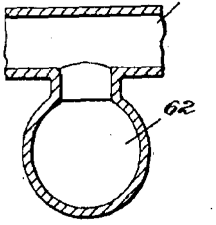


Fig.7.

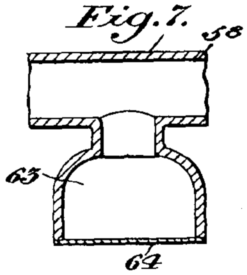


Fig.8.

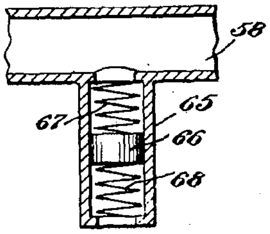
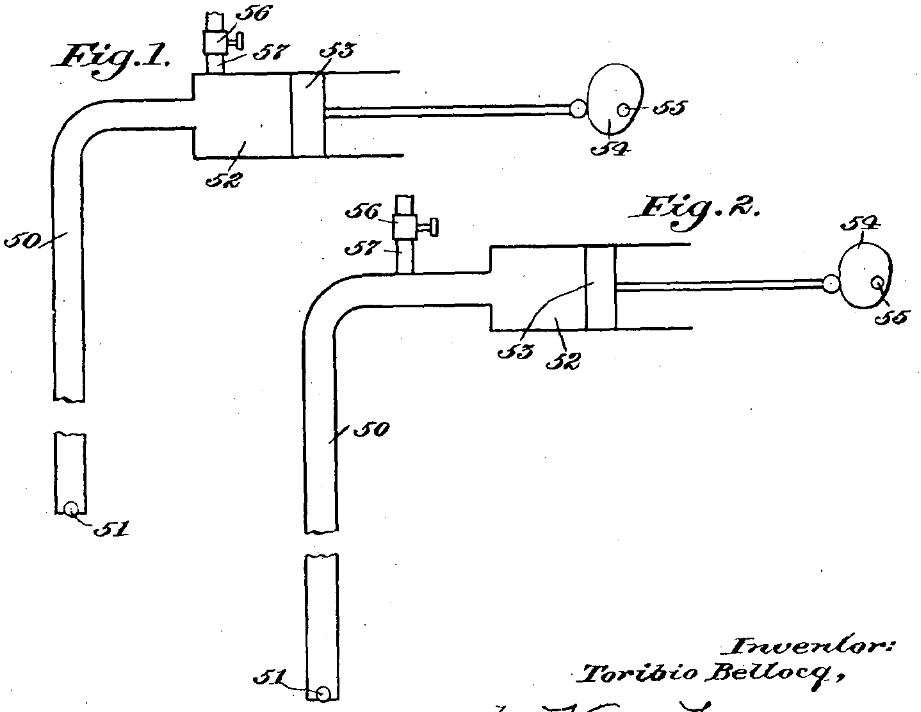


Fig.1.

Fig.2.



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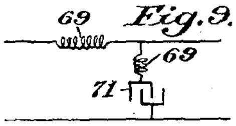
Fig. 9.

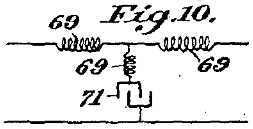
Fig. 10.

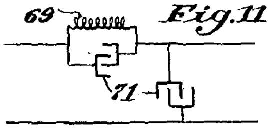
Fig. 11.

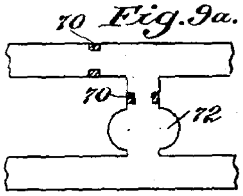
Fig. 9a.

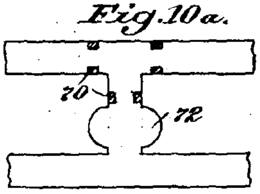
Fig. 10a.

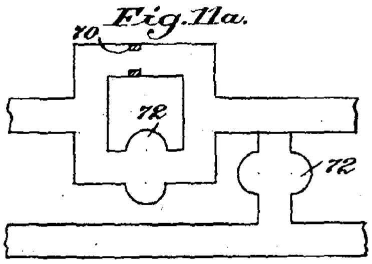
Fig. 11a.

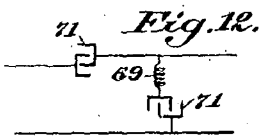
Fig. 12.

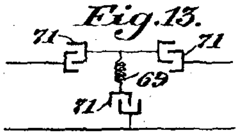
Fig. 13.

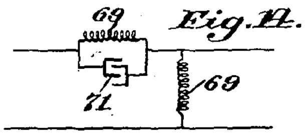
Fig. 14.

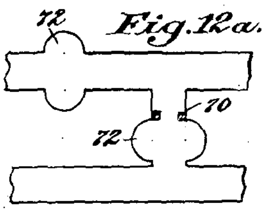
Fig. 12a.

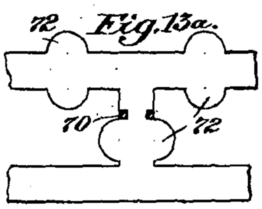
Fig. 13a.

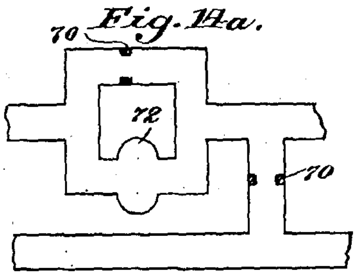
Fig. 14a.

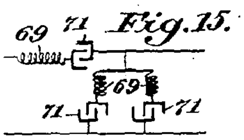
Fig. 15.

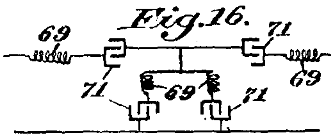
Fig. 16.

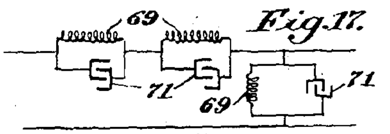
Fig. 17.

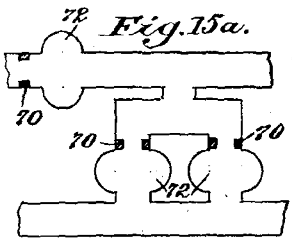
Fig. 15a.

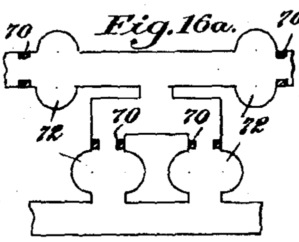
Fig. 16a.

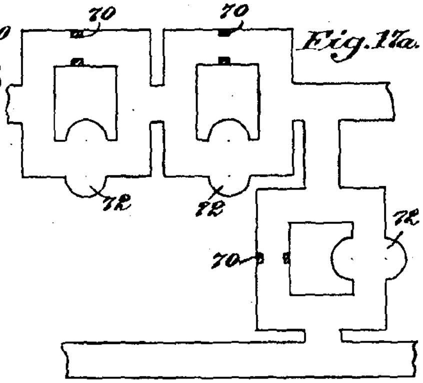
Fig. 17a.

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Fig.18.

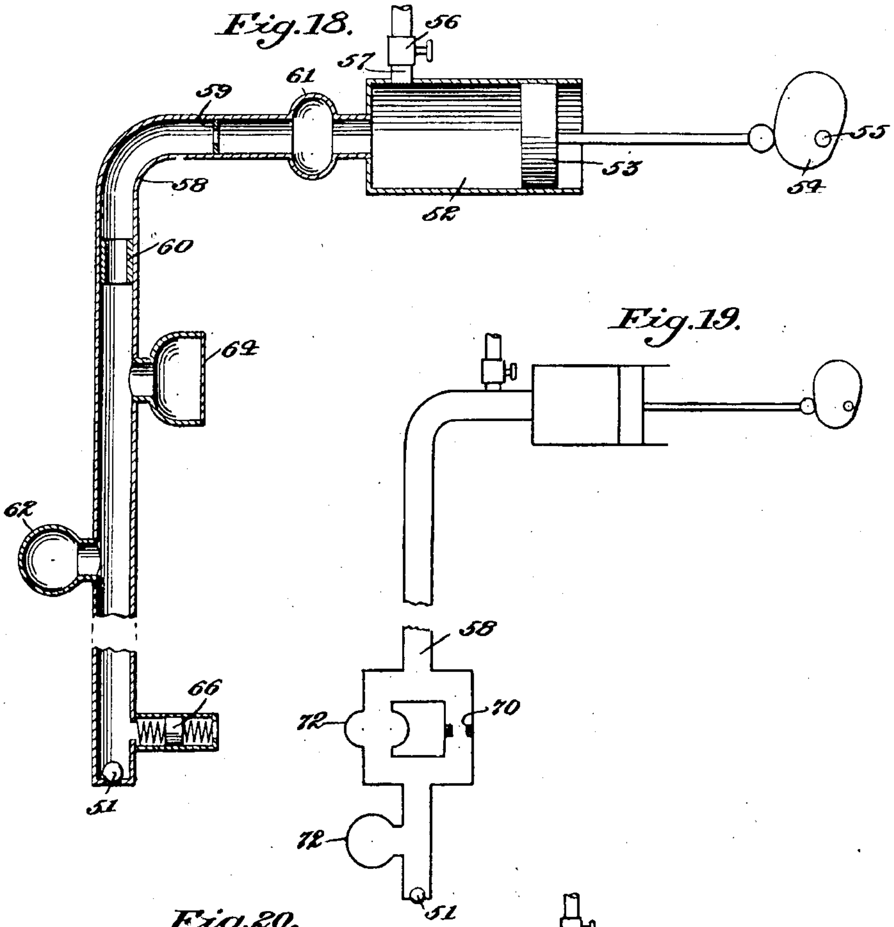
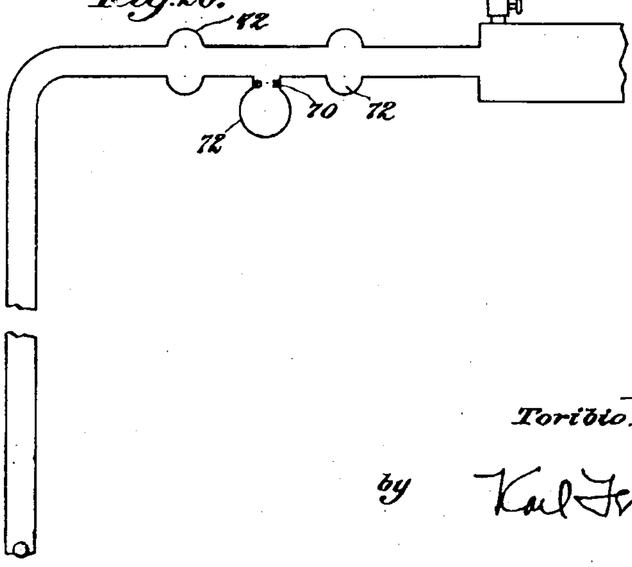


Fig.19.

Fig.20.



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which is applied the devices illustrated in Figs. 3, 4, 5, 6, 7 and 8. Fig. 19 is a diagrammatic illustration of a pumping apparatus such as shown in Fig. 2 to which is applied the apparatus illustrated in Fig. 11*a*. Fig. 20 is a pumping apparatus such as is illustrated in Fig. 1 to which is applied the device illustrated in Fig. 13*a*.

The pumping system in general is illustrated as including a pipe 50 provided with a check valve 51 at its lower end intended to be inserted in the well or other source of supply of liquid to be pumped. At the other end of the pipe is shown a compression chamber or cylinder 52 in which moves a piston 53 controlled by a cam 54 on a suitably operated shaft 55. There is an outlet cock or valve 56 of any suitable type through which the liquid is pumped here shown in an outlet pipe 57. In Fig. 1 the outlet is shown leading directly from the compression chamber or cylinder 52 while in Fig. 2 it is shown as directly connected to the pipe 50 so that the outgoing liquid will not pass through the compression chamber or cylinder. The cock or valve 56 may be open at all times during operation or it may be opened and closed at appropriate times during the operation either manually or automatically.

In Fig. 3 a pipe 58 is shown provided with a narrow annular obstruction 59 which acts as an inertia for the current of waves passing through liquid in the pipe. In Fig. 4 there is shown at 58 a similar inertia 60 which takes the form of an elongated sleeve and so has a different value from the inertia shown in Fig. 3. In Fig. 5 is shown a pipe 58 provided with rigid protuberances 61 which act as a capacity in series with the pipe. In Fig. 6 is shown a pipe 58 which is connected to a closed rigid chamber 62 which when filled with liquid acts as a capacity in derivation. In Fig. 7 is shown a pipe 58 connected with a chamber 63 closed at its outer end by a diaphragm 64. When the chamber 63 is filled with liquid it acts as a capacity in derivation. In Fig. 8 is shown a pipe 58 from which leads a pipe 65 in which is a snugly fitting piston 66 movable against springs 67 and 68. The resiliency of the springs takes the place of liquid resiliency and acts as a capacity in derivation to the pipe 58. These are illustrative only and other forms of inertia and capacity may be employed.

The pipe through which liquid is to be pumped may be horizontal, vertical or inclined or bent. If there is a bend in the pipe the relation between the parts on each side of the bend tends to produce an harmonic of the principal wave and in order to work most efficiently this harmonic should preferably have a node in the elbow at the bend. The elbow thus acts as an inertia.

The entrance or the exit of the liquid pumped may also tend to produce an harmonic four times as long as the length of the liquid which passes through the pipe at each stroke of the compressing device. It is desirable to have this harmonic the same as the harmonic produced by the proportion between the portions of a bent line, or at least an harmonic thereof. The flowing of the liquid through the inlet or the outlet has somewhat the effect of an inertia. The change in size between the pipe carrying the liquid and the cylinder of the compressor at their point of connection has somewhat the same effect as an inertia. The small chamber generally employed at the outlet cock or valve between the compressor piston and the pipe line and likewise the cylinder itself each have somewhat the same effect as a capacity. This chamber may be omitted and the cylinder alone be referred to as the compression chamber.

There are several elements of the apparatus which may be adjusted to change or vary the efficiency of operation including: (1) the size of the compression chamber may be varied; (2) the relative size of the connection between the compression chamber and the pipe may be varied; (3) the cross-section of the pipe through which the liquid is pumped may be varied; (4) the speed or timing of the piston may be varied: (5) the size or cross-section of the piston or its length of stroke may be varied (6) the size or opening of the check valve through which liquid enters the system may be varied; or (7) the size and suddenness of the opening of the outlet may be varied. In order to procure proper and appropriate adjustments or regulation of the system any or all of the various elements may be appropriately and correspondingly varied. The quantitative effect of variations in the different elements of the apparatus is not the same.

A difference in operation is produced when the outlet is in the compression chamber from that produced when the outlet is on the pipe. In the first arrangement increasing the size or opening of the check valve or in the second arrangement decreasing the size or the opening of the check valve effects a variation in the same direction as (1) increasing the size of the compression chamber (2) decreasing the relative size of the connection between the chamber and the pipe (3) decreasing the cross-section of the pipe through which the liquid is pumped (4) increasing the speed or decreasing the timing of the piston (5) increasing the size or cross-section of the piston or its length of stroke (6) increasing the size of opening of the outlet. This indicates the direction of change to be made to compensate for any change in the apparatus. If the outlet on the pipe is removed farther from the connection between the pipe and the compression chamber or if inertia associated with the outlet is moved farther from the connection between the pipe and compression chamber the effect is of the same character as would be an enlargement of the compression chamber. It is generally desirable, however, to put the outlet, the inertia and the connection between the pipe and the compression chamber as close together as possible because when they are separated one from another there is no longer merely the effect of a chamber assembled at one place but the effect of a chamber like a piece of pipe with different inductions in each extremity. With such an arrangement it is more difficult to calculate the exact value of such a chamber. Sometimes, however, it is found desirable to use such an extended chamber despite the difficulty in calculating its characteristics.

It may happen that when the pipe is very long the wave will be deformed as it proceeds through the liquid. It may then be desirable to put additional or new capacities, inertias, resistance or leakage in one or several parts of the apparatus to prevent this deformation of the current. Somewhat similarly in telegraphic and telephonic lines traveling great distances means are provided to prevent deformation of the circuit wave such as are provided by the system of Puppini or others.

It may be desirable to choose a vibration or wave of such a character that the liquid will vibrate with the natural vibration produced by gravity. For example a compressor may be so operated as to cause an artificial acceleration during its stroke corresponding to the natural acceleration produced by gravity on the liquid freely falling in the system. When this natural vibration is not convenient for the quantity of liquid it is desired to pump the constants of the line may be appropriately changed to produce the desired effect.

When the pipe is large enough or the quantity of the liquid pumped is small in relation to the size of the pipe, the resistance will be small and for practical purposes may be neglected. It is desirable that the leakage in the system, be reduced to a minimum so that the only value of leakage to be considered may be that produced by the liquid pumped through the outlet. Entrance of liquid into the system may be considered as negative leakage.

The compression waves in the liquid of the pumping system are somewhat analogous to sound waves. Therefore, elements which modify the waves in the liquid of the pump may for convenience be referred to as acoustic elements and it is to be understood that when the term acoustic or hydraulic is employed herein it relates to an element modifying the compression waves in the liquid more or less in accordance with the laws of sound waves in the liquid. This may be a convenient form of expression and it may aid in understanding the present invention to bear in mind the analogy between the compression waves in the liquid in the pumping system and sound waves.

Acoustic inertia or self induction may be an apparatus having only inertia which may be formed for instance by a reduction of a section of the pipe. Likewise acoustic capacity may be a chamber communicating with the pipe and filled with the liquid. Acoustic inertias, capacities, resistances and leakages may be combined to form acoustic filters analogous to electric filters and their effect on the compression waves in the pumping system may be analogous to their effect on sound waves. As in electricity such acoustic filters in the pumping system may be referred to as low pass filters, high pass filters and band pass filters. The low pass filters may let pass or produce vibrations of a frequency lower than a certain speed or may so convert other vibrations received. The high pass filters may let pass or may produce vibrations of a frequency higher than a certain speed or may so convert other vibrations received. The band pass filters may let pass or may produce vibrations of a frequency between two certain speeds, or may so convert other vibrations received. All of these filters may stop or not let pass other vibrations. Acoustic filters may be arranged in accordance with the same laws as electric filters. It may be easier to make filters occupying a very short space in relation to the wave length. It is possible, however, to employ other longer filters but the exact determination of their effect may be more difficult.

The analogies between electric filters and acoustic filters exist not only in words but they are true physical analogies. The phenomena of the propagation of an electric current and of sound waves and of compression waves in liquid are controlled by the same laws. In dealing with these waves the combination of acoustic inertia, capacities, resistance and leakages accomplish similar results as are accomplished by combinations of similar elements in electric circuits. In order to facilitate determining the effect of a filter it is desirable that the two extremities of the filter be of the same value. It is possible, however, to employ filters having extremities of different values. In such an arrangement the difference in the two extremities may produce an effect of itself. The ideal filter is one which without substantial loss of energy will let pass the frequency which it is desired to employ in the apparatus and at the same time stop other frequencies. Generally filters of this character cannot be completely realized in practice, but it is desirable to employ filters approaching this ideal as nearly as possible. A general filter may be employed or several filters may be employed in a single system being associated or combined with similar or different filters.

Generally electric circuits employ two wires or lines and the induction, capacity, resistance and leakage may be installed in one or both or between them in series, derivation, or in shunt. In liquid systems, especially pumping systems, usually there is but a single line or pipe but the inertia, capacity, resistance and leakage may be placed in series or in shunt on the line or in derivation on the line in which case the effect is as if the element in derivation communicated at one side with the line and at the other side with a ground analogous to the grounding of an electrical device.

The effect of filters put in derivation may be varied, for example by putting a capacity in derivation made not of a closed chamber filled with the same liquid but of a closed chamber filled with another liquid of different density or elasticity, separated by a membrane from the other liquid or by a chamber filled by the same liquid but closed by a membrane which separates the liquid from the air. The capacity thus is to be considered as not between the line and the ground but between the line and another artificial ground. The natural vibration of the liquid will be different in the different cases. This chamber filled with the different liquid or provided with a membrane may be replaced by a cylinder having a floating piston maintained by springs which make the piston work like a membrane. The introduction of any of these different apparatus having vibration characteristics different from the natural vibrations of the line has the effect of changing the constants of the line and may be employed in a manner similar to the more usual type of filters. Any other suitable form of capacity, inertia, resistance or leakage may be employed.

For installations operated under substantially constant conditions a filter of exact characteristics may be employed, but for other installations less exact filters may be employed.

In Figs. 9, 10 and 11 are shown examples of characteristic electrical low pass filters. In Figs. 9*a*, 10*a* and 11*a* are shown similar characteristic low pass acoustic filters applied to liquid systems. The arrangement in Fig. 9*a* corresponds to that in Fig. 9; the arrangement in Fig. 10*a* corresponds to that in Fig. 10, while that in Fig. 11*a* corresponds with that in Fig. 11. In each instance the electrical inertias 69 are replaced by acoustic inertias 70 while the electric capacities 71 are replaced by acoustic capacities 72. Figs. 12, 13 and 14 illustrate types of electric high pass filters and Figs. 12*a*, 13*a* and 14*a* illustrate types of acoustic high pass filters corresponding respectively to the electric filters illustrated in Figs. 12, 13 and 14. Figs. 15, 16, and 17 illustrate diagrammatically electric band pass filters and Figs. 15*a*, 16*a* and 17*a* illustrate characteristic acoustic band pass filters applied to liquid systems corresponding respectively to the filters illustrated in Figs. 15, 16 and 17. In all of these figures the electrical inertias 69 correspond to the acoustic inertias 70 and the electric capacities 71 correspond to the acoustic capacities 72. The filter arrangements here illustrated are selected as types only. These and other appropriate electric filters may be followed with like effect and the corresponding acoustic filters inserted at appropriate points in liquid systems with corresponding results and the term filter is used to include all such devices as rectify, transform, alter, or modulate the currents of the compression waves set up in the liquid in the pumping system.

In Fig. 18 is shown a simple pumping system such as shown in Fig. 1 but modified by being associated with the filters or wave modifying devices illustrated in Figs. 3 to 8 inclusive. The capacities 61, 62, 64 and that having the piston 66 are associated with inertias 59 and 60. As illustrated the inertia 59 and the capacity 61 are shown relatively near to the pump cylinder 52 and the other wave modifying devices are distributed throughout the line 66 being very near to the inlet valve 51. 59, 60 and 61 are in series while 62, 64 and 66 are in derivation. This specific arrangement is not essential nor is it essential that all of these devices be incorporated in a single system. In general a single device may be sufficient. For instance the inertia 59 might be employed and the devices 60, 61, 62, 64 and 66 omitted or any two or more of them might be omitted. Likewise the particular position of these various devices in the system is not essential. For instance the device 66 which is shown as near the check valve 51 might be installed where the device 61 is illustrated near the pump cylinder 52 or any other desired arrangement and location of any or all of these devices may be made. Instead of assembling in a single system a plurality of different wave modifying devices as illustrated in Fig. 18 a plurality of similar devices might be employed when more than one is desired and the location of all of the devices is interchangeable. When apparatus such as is illustrated in Fig. 1 is operated the wave produced in the liquid will have certain characteristics and for the conditions of operation it may be that that wave is satisfactory and efficient. It may be, however, that for the conditions under which the apparatus is operating the wave produced by apparatus such as illustrated in Fig. 1 will not have the desired characteristics and may not be effectively efficient to produce the desired pumping effect. By adding to the apparatus shown in Fig. 1 any one or more of the devices illustrated in Figs. 3 to 17 the characteristic of the wave may be modified so as to produce the desired effective efficient result. Thus the wave set up in such an apparatus as Fig. 18 will have characteristics different from the wave set up in such an apparatus as Fig. 1 and the addition or removal of any one of the impedance or capacity devices illustrated in Fig. 18 will correspondingly modify the wave.

In Fig. 19 is shown assembled in such a pump as is shown in Fig. 2 a filter such as illustrated in Fig. 11*a* and in Fig. 20 such a filter as illustrated in Fig. 13*a* is illustrated in connection with such a pump as shown in Fig. 1. As indicated, when the arrangement is applied to a pumping system there may be present only one pipe line the other pipe being omitted and replaced by an artificial ground similarly to the arrangement by which a two wire electric system may be employed with only one wire and a ground. In the pumping system the effect of a ground may be procured by closing the connection to the omitted pipe. Likewise the filters illustrated in Figs. 9*a*, 10*a*, 12*a*, 14*a*, 15*a*, 16*a* and 17*a* or any grouping or combinations of them or other filters may be assembled into a single apparatus. The arrangement illustrated in Fig. 19 is in part a shunt arrangement and, incorporating a low-pass filter, will modify the waves in the pumping system and allow only those waves having frequencies lower than a certain desired speed or it may convert to such speed other vibrations in the system. Similarly the filter associated with the pumping system in Fig. 20 being a highpass filter will modify the waves in the pumping system and allow only those waves having frequencies higher than a certain desired speed or it may convert to such speed other vibrations in the system.

The devices shown are only preferred embodiments of the invention, but any other suitable devices may be used for the same purposes.

The weight of the check valve or the strength of springs sometimes controlling it which change the character of its opening change the value of the inertia produced by its opening. The check valve being inertia and so a filter element may be omitted and the other elements of the apparatus adjusted accordingly. When more than one pump is in the pipe line each pump may act as check valve or inlet for the succeeding pump.

Variations in the speed of the piston during a stroke are of importance. When the piston is moved by a crank shaft the speed increases from zero to maximum speed during the first half of the time of the entrance of the piston (that is, the first quarter of a complete cycle) and produces an acceleration in the lifting of the liquid. In the second half of the time of entrance of the piston (that is, the second quarter of a complete cycle) the speed of the piston decelerates going from its high speed to zero speed. Thus the liquid tends to be lifted with more strength when the piston is entering during the first half of its forward stroke and tends, because of the deceleration, to be lifted with less strength in each unit of time when the piston is entering during the second half of its forward stroke. In general under these conditions it is better to open the outlet valve when or before the piston reaches the half point of the inward stroke. Sometimes it may be better to open the valve at the end of the entrance stroke of the piston or somewhat before. When a proper cam is employed to operate the piston, the piston speed increases throughout the entire advance movement so the same effect may sometimes be produced by a cam which gives only one-half the length of movement to the piston as would be given by a corresponding crank. When such a cam is used only only-half as much liquid may be forced out of the piston cylinder as with a crank and hence at each stroke less liquid need go into the cylinder to keep it full in order to get as much liquid output from the apparatus. With the most efficient adjustment of the apparatus, during the acceleration of the piston liquid is lifted, that is to say, the liquid may be moving from the check valve to the compressor. If the valve is opened at this moment the liquid which is already moving upward, will tend to be lifted with more efficiency since it is not obliged to change its direction.

The suddenness of opening the outlet being the principal element in the transformation of the compression waves should be especially considered and arranged to cooperate with the rest of the system.

In certain cases it may be desired to produce a certain pressure in the apparatus at the time the outlet is opened as in this connection it seems that the liquid pumped going out with a certain pressure will carry with it enough energy to lift the liquid higher than the apparatus. The apparatus working in this form may work somewhat like a combination of the system employing the open cock at the outlet and the system opening and closing the valve, retaining thus the advantages of both systems.

The conditions involved in the use and application of filters indicate the important effect which may be produced by small defects in the apparatus such as leakage, holes or couplings or connections which allow liquid to pass out, these defects being analogous to the bad effect produced in electrical circuits by bad contacts or by other defects in different parts of the apparatus, etc. The piston may work either with the cylinder horizontal or vertical or inclined, or any other suitable means may be employed to produce variations in compression in the liquid to set up the compression waves. The pipe through which liquid flows may be either horizontal or vertical or inclined, and it may be curved, bent or straight. All of these arrangements may be referred to as pumps, and the term pumping is used as including moving, transporting or conveying in vertical or horizontal or inclined directions or a combination of any of them, and the liquids so pumped may be simple liquids or may carry more or less solid matter or gases.

I claim as my invention:

1. The method of varying and adjusting the efficiency of pumping apparatus comprising a pipe carrying liquid an inlet and an outlet therefor and means for causing compression waves to pass through the liquid and thereby cause movement of liquid through the apparatus, comprising associating and communicating a wave filter with the pipe so as to modify the waves in the liquid.

2. Pumping apparatus comprising a pipe carrying liquid, an inlet and an outlet therefor, means for causing compression waves to pass through the liquid and thereby cause movement of liquid through the apparatus, a wave filter associated and communicating with the pipe so as to modify the waves in the liquid.

3. Pumping apparatus comprising a pipe carrying liquid, an inlet and an outlet therefor, means for causing compression waves to pass through the liquid and thereby cause movement of liquid through the apparatus, inertia, capacity and leakage associated and communicating with the pipe so as to modify the waves in the liquid.

4. Pumping apparatus comprising a pipe carrying liquid, an inlet and an outlet therefor, means for causing compression waves to pass through the liquid and thereby cause movement of liquid through the apparatus, inertia associated and communicating with the pipe so as to modify the waves in the liquid.

5. Pumping apparatus comprising a pipe carrying liquid, an inlet and an outlet therefor, means for causing compression waves to pass through the liquid and thereby cause movement of liquid through the apparatus, capacity associated and communicating with the pipe so as to modify the waves in the liquid.

6. Pumping apparatus comprising a pipe carrying liquid, an inlet and an outlet therefor, means for causing compression waves to pass through the liquid and thereby cause movement of liquid through the apparatus, inertia and capacity associated and communicating with the pipe so as to modify the waves in the liquid.

7. Pumping apparatus comprising a pipe carrying liquid, means for causing compression waves to pass through the liquid, whereby the waves cause liquid to flow through the apparatus, and an auxiliary wave filter associated and communicating with the pipe so as to modify the waves.

8. Apparatus for pumping liquids comprising a check valve in the liquid supply, a compressor at the point of delivery including a cylinder and a piston and an outlet pipe, a pipe filled with liquid extending from the valve to the compressor, means for reciprocating the piston rapidly through a short stroke whereby the compression of the liquid produced by the piston is varied and a wave filter associated and communicating with the last named pipe which when altered varies the output of the pump.

9. An apparatus for pumping liquid comprising a pipe filled with liquid leading to the liquid supply, a check valve in the pipe in the liquid supply, a compressor cylinder at the other end of the pipe, an open outlet from the cylinder, a piston to alternately compress the liquid in the pipe and release the pressure, and a wave filter associated and communicating with the pipe.

10. Apparatus for pumping liquid comprising means for alternately compressing a body of liquid in a pipe and releasing the pressure to set up waves in the liquid, an outlet for the pipe whereby the waves cause liquid to enter and flow through the pipe and auxiliary wave filters associated and communicating with the pipe which when altered vary the liquid flow through the pipe.

11. Apparatus for pumping liquid comprising means for alternately compressing a body of liquid in a pipe, separate means for alternately releasing the pressure, an outlet for the pipe whereby the variations in compression cause the liquid to enter and flow through the pipe, and an auxiliary wave filter for varying the flow.

12. Apparatus for pumping liquids comprising a pipe leading to the liquid supply filled with liquid, a check valve in the pipe within the liquid to be moved, an outlet for the liquid, means for compressing the liquid in the pipe, separate means for suddenly releasing the pressure, and an auxilary wave filter for varying the flow of the liquid.

13. Apparatus for pumping liquids comprising a pipe leading to the liquid supply and filled with liquid, a check valve in the pipe within the liquid supply, an outlet for the liquid, means to set up a contemporaneous plurality of series of compression waves in the liquid in the pipe to operate the check valve and move the liquid, and an auxiliary wave filter for varying the flow of the liquid.

14. Apparatus for pumping liquids comprising means for alternately compressing a body of liquid in a pipe and releasing the pressure, an outlet for the pipe, whereby the variations in

compression cause liquid to enter and flow through the pipe, and an enlargement connected with the pipe modifying the pumping effect.

15. Apparatus for pumping liquids comprising means for alternately compressing a body of liquid in a pipe and releasing the pressure, an outlet for the pipe, whereby the variations in compression cause liquid to enter and flow through the pipe, and an enlargement in the pipe modifying the pumping effect.

16. Apparatus for pumping liquids comprising means for alternately compressing a body of liquid in a pipe and releasing the pressure, an outlet for the pipe, whereby the variations in compression cause liquid to enter and flow through the pipe, and an obstruction in the pipe modifying the pumping effect.

17. Apparatus for pumping liquids comprising means for alternately compressing a body of liquid in a pipe and releasing the pressure, an outlet for the pipe, whereby the variations in compression cause liquid to enter and flow through the pipe, and an obstruction and an enlargement in the pipe modifying the pumping effect.

18. Apparatus for pumping liquids comprising means for alternately compressing a body of liquid in a pipe and releasing the pressure, an outlet for the pipe, whereby the variations in compression cause liquid to enter and flow through the pipe, and an obstruction and an enlargement in a passageway connected with the pipe modifying the pumping effect.

19. The method of regulating the output of a pumping apparatus comprising means for alternately compressing a body of liquid in a pipe, separate means for alternately releasing the pressure and an outlet for the pipe, whereby the variations in compression cause the liquid to enter and flow through the pipe, consisting of inserting and regulating, varying and/or adjusting auxiliary and additional wave filter elements.

20. The method of regulating the output of apparatus for pumping liquid comprising means for alternately compressing a body of liquid in a pipe, separate means for alternately releasing the pressure, an outlet for the pipe whereby the variations in compression cause the liquid to enter and flow through the pipe, and an auxiliary wave filter for varying the flow, consisting of regulating, varying and/or adjusting the auxiliary wave filter.

21. The method of regulating the output of apparatus for pumping liquids comprising a pipe leading to the liquid supply and filled with liquid, a check valve in the pipe within the liquid supply, an outlet for the liquid, means to set up a contemporaneous plurality of series of compression waves in the liquid in the pipe to operate the check valve and move the liquid, and an auxiliary wave filter for varying the flow of the liquid, consisting of regulating, varying and/or adjusting the auxiliary wave filter.

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Feb. 25, 1941. R. H. DICKINSON 2,232,678

PUMP FOR LIQUIDS

Filed March 17, 1938

Fig. 1.

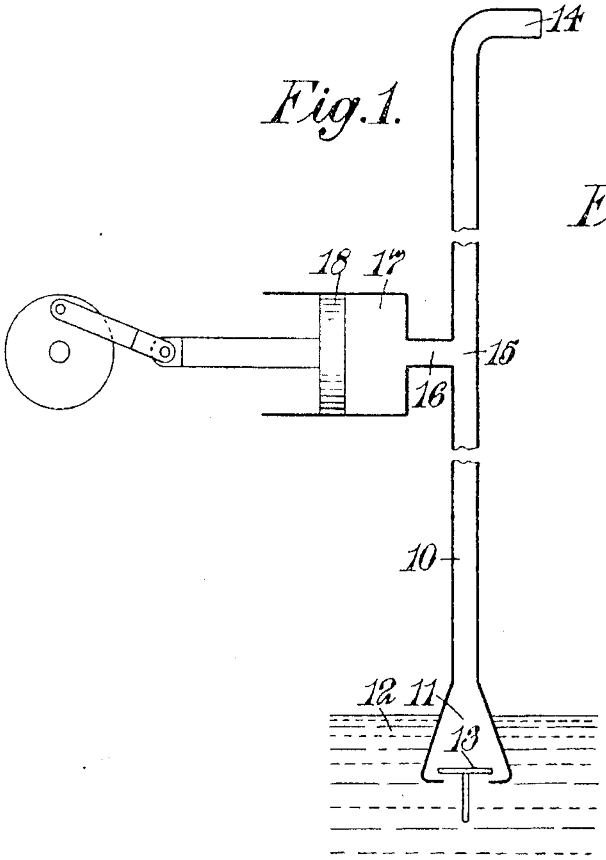


Fig. 2.

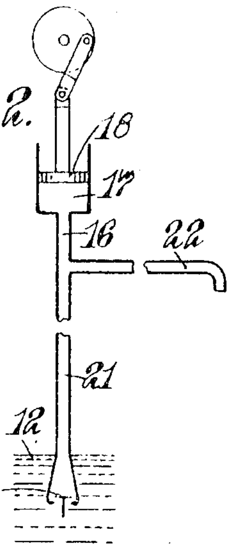


Fig. 3.

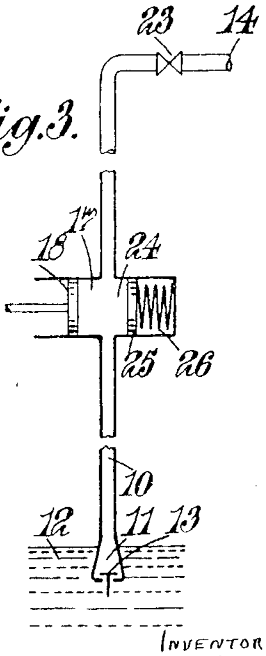
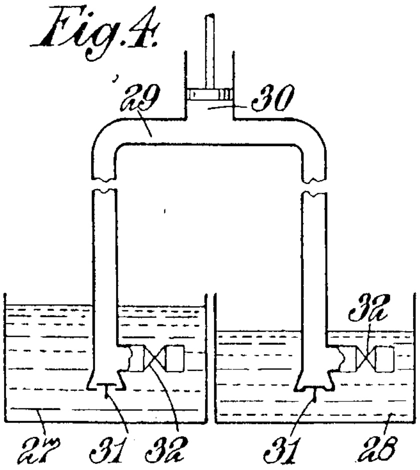


Fig. 4.



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2,232,678

PUMP FOR LIQUIDS

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Application March 17, 1938, Serial No. 196,499
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4 Claims. (Cl. 103—75)

This invention is for improvements in or relating to pumps for liquids, and has for its object to provide an improved construction and arrangement of a pump which is extremely simple in its mechanical details, and is capable of raising liquids efficiently, even from a depth below the mobile portion of the pump greater than the barometric column, and of delivering it at any required pressure.

Pumps made according to this invention comprise as their essential features a submerged inlet pipe provided with a non-return inlet valve, situated as hereinafter described, an outlet pipe, communicating with the inlet pipe, and means for intermittently applying pressure at their junction, which point for convenience is hereinafter referred to as the "pressure chamber."

As a result of such pressure the column of liquid above the pressure chamber is set in motion so as to separate, or tend to separate, it from that below the pressure chamber. This pressure also tends to compress the inlet column and to transfer energy to the inlet part of the system, thereby causing expansion and/or extension of the piping, and to a certain amount, compression of the liquid itself. On the removal of the pressure on the upper surface of the liquid in the inlet part this energy, or a portion thereof, is utilised to raise the liquid in such part, and thereby enable the foot valve to open (even in cases where the inlet column is of greater height than the barometric column) and further liquid to be admitted. It is therefore an essential feature of this invention that the foot valve should be situated at a sufficient distance below the pressure chamber to permit of a considerable amount of energy being stored in the inlet portion of the system, and preferably this valve is situated at the bottom of the inlet pipe, as any portion of such pipe below the foot valve is not available for storing energy.

We also prefer to arrange the outlet pipe, by which we mean that portion of the system which is above the pressure chamber, so that there is a substantial head of liquid therein, as this facilitates the imparting of energy to the inlet column, since it enables greater effective pressure to be developed thereon. In all cases the head of liquid must be adequate to enable sufficient pressure to be applied to the surface of the inlet column so as to transfer thereto the energy necessary to set this column in upward motion, and thereby enable liquid to enter through the foot valve. It is necessary that the pipes forming the inlet part of the system, and/or the liquid, should possess sufficient elasticity for storing the necessary energy, but we have found that with the comparatively small pumps with which we have experimented up to now the ordinary cast iron, steel or earthenware pipes are satisfactory, and we have also worked with rubber hose.

In order to start the pump it is necessary that it should be primed, i. e., there should be a sufficient column of liquid above the level of the pressure chamber.

Preferably pressure is applied by means of a piston reciprocating in a cylinder which is in free communication with the inlet and delivery pipes.

The invention contemplates the provision of an improved construction and arrangement of pump of the aforesaid type, in which all valves except the inlet valve are iliminated, and which operates on a cycle that is independent of its speed; it is thereby distinguished from known pumps of the said type which operate by setting up resonant pressure waves in the system and also from those which operate on a cycle determined by the operation of their valves.

In the accompanying drawing,

Figure 1 is a purely diagrammatic illustration of one embodiment of the invention.

Figure 2 is a similar illustration of a modified arrangement of inlet pipe and delivery pipe.

Figure 3 is a similar illustration showing means for controlling the delivery of the pump, and

Figure 4 is a similar illustration showing the novel pumping system adapted for transferring liquids in either of two directions.

Like reference characters indicate like parts throughout the drawing.

Referring first to Figure 1 which shows diagrammatically a pump system according to the present invention in its simplest form, a pipe 10 has its lower end 11 submerged in an intake source 12, and provided with a non-return inlet valve 13; the upper end of the pipe 10 constitutes the delivery point. At any convenient point 15 between its ends, a connection 16 provides communication with a cylinder 17 having a piston or plunger 18 reciprocated in it by any convenient driving mechanism. No valves need be provided in the cylinder 17 or in its connection with the pipe 10, the sole function of this cylinder and piston being to exert pressure intermittently on the contents of the pipe 10. It will be seen therefore that many other devices than a cylinder and piston may be used in its place.

The connection 16 to the pump cylinder may be situated at any convenient point in the length of the pipe 10 and in the case of a deep-lift pump it may be situated above the height of a barometric column of the liquid which is being pumped. The pipe 10 is continuous from its inlet end 11 to its delivery end 14, and the portion of it between the inlet end 11 and the connection 15 constitutes the inlet pipe, and the portion of it between the connection 15 and the delivery end 14 constitutes the delivery pipe and these names will be used to designate these two parts hereinafter.

The operation of this pump is as follows:

The whole system is primed with liquid as above described, and preferably the connection 16 and cylinder 17 are also filled with liquid when the piston 18 is at the outer end of its stroke. The non-return inlet valve 13 is held closed by the pressure of this column of liquid. When the piston is forced inwards it exerts a pressure on the liquid in both the inlet and delivery pipes and since the non-return valve 13 is closed no liquid can escape from the inlet pipe and the liquid in the pipe itself is thus subjected to the full pressure exerted by the piston, and the inlet part of the system with the liquid in it is strained so that the piping is expanded and the liquid is compressed according to their respective physical characteristics.

Simultaneously, the application of the pressure by the piston 18 to the liquid in the delivery pipe causes it to be discharged therethrough, and preferably the dimensions of this pipe are so selected that the liquid is given a high velocity. Conveniently, this is effected by employing a delivery pipe of small diameter relatively to the cylinder of the pump.

Where the delivery end 14 is open, the pressure in the system which can be generated by the piston 18 depends upon the mass of liquid in the delivery pipe and the acceleration which is imparted to it, and it is desirable that this pressure should be high and this is effected by ensuring either that there is a large mass in the delivery pipe, or that it is given a very high velocity, or both.

When the liquid in the delivery pipe has attained its maximum velocity the piston is approaching, or is at, the inner end of its stroke depending upon the mechanical arrangements used for operating the piston; if it is driven through a crank and connecting rod it will have an approximate harmonic motion and the maximum velocity of the liquid in the delivery pipe will occur at or shortly after the middle of the inward stroke of the piston. As soon, however, as the piston slows down or stops, the liquid in the delivery pipe having attained its maximum velocity will tend to continue in motion, by virtue of its inertia so that it will separate or tend to separate from the liquid in the cylinder and in the inlet part of the system. As the pressure on the inlet part of the system is relieved the energy which has been stored in the inlet part of the system is released and the liquid in the inlet part of the system is thereby set in motion towards the delivery pipe, and by being lifted off the inlet valve it relieves the pressure thereon so that an inflow of liquid takes place due to the atmospheric pressure outside; it will be seen therefore that when the piston is ready to start on its next inward stroke, some liquid has been discharged through the delivery pipe from the open end 14 thereof, some of the liquid in the inlet pipe has been transferred to the delivery pipe and to the pump cylinder, and additional liquid has entered the inlet pipe through the non-return valve 13. On the next stroke the cylce of operation is repeated and this continues so long as the piston is kept in motion so that liquid is lifted from the supply 12 and delivered from the outlet 14.

It will be seen that special advantages arise from this characteristic of the present invention as for example in the case of pumping a liquid which contains a gas in solution. This is troublesome with an ordinary type of suction pump since the gas tends to come out of solution, whereas in the present system the presence of gas in solution renders the liquid more highly compressible so that more energy can be stored in it, and the fact that the liquid is subjected to pressure prevents the gas from coming out of solution.

It also lies within the scope of the invention to provide special means in the inlet part of the system whereby energy may be resiliently absorbed, it being appreciated of course that such special means must always be capable of withstanding the pressure to which the system may be subjected in its operation.

In cases where comparatively small lifts are required on the inlet side, the inlet pipe may be made of a suitable resilient material such as stout rubber hose; alternatively, a metal pipe may for example be lined with rubber or other resilient material which will easily absorb and return energy to the system as required, and it will be appreciated that this is useful as for example when pumping corrosive liquids.

The following is an example of an actual construction of a pump used in practice.

The piston was of 2 ins. diameter and had a 4 ins. stroke and was actuated by a 1 H. P. motor. The inlet pipe was of mild steel of 1 inch diameter approximately 66 feet long. The water level was 46 feet below the bottom of the pump cylinder which was disposed vertically, and the foot valve was 20 feet below water level. The delivery pipe was 100 ft. long, the first 30 feet being of 1½ ins. diameter and the remainder of 3 ins. diameter, and the outlet point of such delivery pipe was 50 feet above the pressure chamber. When working at a speed of 340 strokes per minute, a delivery of 2,400 gallons per hour could be obtained, but the pump could be run at lower speeds absorbing less power.

It will of course be appreciated that the outlet and inlet pipe may form one continuous whole so long as there are efficient means for applying the intermittent pressure and that the term "pressure chamber" as here used covers any arrangement for the application of such pressure.

It is not essential that the piping should be arranged as shown in Figure 1, and Figure 2 shows a different arrangement in which the inlet pipe 21 with the non-return valve 13 is vertical and the delivery pipe 22 is horizontal. In this case the cylinder 17 is shown as arranged vertically and the connection 16 to the junction between the inlet and delivery pipes is apparently a continuation of the inlet pipe 21. Although this arrangement gives the appearance that the delivery pipe is situated on the inlet side of the pump, it is not so in fact because the delivery pipe is constituted as before by that part of the pipe between the connection to the cylinder and the open delivery end.

It will be appreciated therefore that any disposition of inlet and delivery pipes may be used, and the cylinder and piston may be situated at any point provided it communicates with the junction of the inlet and delivery pipes so that the pressure can be applied intermittently to them. Preferably, this connection will be filled with liquid, as will the cylinder, in order that the requisite pressures can be obtained, but in some instances it may be possible to provide an air-cushion between the pump piston and the liquid in the main part of the system.

It will be appreciated that in the system so far described the only valve present in the system is the automatic non-return inlet valve **13**, so that the rate of delivery of fluid by this pump is dependent solely on the speed of operation of the piston **18**. If it is desired to provide a manual valve control this may be effected as shown diagrammatically in Figure 3. A manually operated stop valve **23** is shown at the delivery end **14** of the system, and in order to prevent pressure building up in the system when this valve is closed and the piston is left running a cylinder **24** having a capacity equal to that of the cylinder **17** is provided in direct communication with the inlet and delivery pipes, and a piston **25** in this cylinder is provided wth a controlling spring **26** normally holding the piston **25** at the inner end of its stroke. This spring is of such stiffness that in normal operation of the system it does not move, but when the valve **23** is closed the pressure which developed on the inward movement of the piston **18** moves the piston **25** so that the liquid is merely transferred between the cylinders **17** and **24** and the pressure on the inlet part of the system is never relieved so that no fresh liquid enters the system and therefore the pressure does not build up.

If the valve **23** is partly opened a restricted delivery is obtained, but the pressure on the inlet part of the system is not released to the same extent and therefore a correspondingly reduced intake of liquid through the valve **13** takes place.

Sometimes it is required to transfer liquids between two points in either direction and the pumping system according to the present invention lends itself to such an arrangement. This is diagrammatically illustrated in Figure 4 in which there are shown two tanks **27**, **28**. The pipe **29** between them is connected to a pumping cylinder **30** in the manner above described, and each end of the pipe is provided with an automatic non-return valve **31** and with a stop valve **32** which can be opened or closed at will, the stop valve opening into the inlet pipe above the inlet valve **31**. If one valve **32** is closed and the other is opened, the second valve **31** is thereby rendered inoperative and pumping takes place, in the manner previously described, from the tank in which the stop valve **32** is closed into the tank in which the stop valve is open. If it is desired to reverse the direction of pumping all that is necessary is to close the second valve **32** and open the first valve **32**; whichever of the non-return valves **31** is rendered operative by the closing of its stop valve determines that that end of the pipe constitutes the inlet and the other end of the pipe the outlet or delivery end.

Finally, it is to be pointed out that the drawing accompanying this specification is purely diagrammatic and is not intended to limit in any way the mechanical design or arrangement of the system, and any suitable types of valves and any other kind of pump capable of intermittently producing pressure may be used, the only requisite being that a sufficient interval of time should be provided between the pressure impulses to allow of the transfer of liquid in the manner hereinbefore described and that the valves shall effectively accommodate such flow.

The present invention thus provides an improved construction and arrangement of pump of the aforesaid type in which all valves except the inlet valve are eliminated, and which operates on a cycle that is independent of its speed; it is thereby distinguished from known pumps of similar types which operate by setting up resonant pressure waves in the system, and also from those which operate on a cycle determined by the operation of their valves.

I claim:

1. The method of raising liquids from a depth unrelated to the barometric column of said liquid, which comprises establishing a column of liquid in a suitable inherently resilient containing system having non-return inlet valve means adjacent the bottom thereof, establishing another column of liquid of considerable length beyond and in communication with the upper end of said first named column, and in a suitable containing means comprising a free and unobstructed delivery system, causing a forward acceleration of the liquid in the delivery system and simultaneously causing a compression impulse in the first named or inlet system by applying pressure at the point of junction of said two bodies of liquid, thus tending to separate the two bodies of liquid at said point, storing the energy of the latter impulse in the first named or inlet system by the inherent resilience thereof, and then releasing the stored energy by removing said pressure, and causing the body of liquid in the inlet system to follow the still moving body of liquid into the outlet system, and thus admitting more liquid through the inlet valve means to replace the portion of the liquid which has moved into said delivery system.

2. A pumping system for liquids comprising an inlet pipe with a non-return inlet valve at its intake end, a normally open unobstructed delivery pipe having one end in direct communication with the outlet end of the inlet pipe, the inlet pipe being of considerable length to provide sufficient inherent resilience and expansion for the storage of energy resulting from intermittently applied pressure, the delivery pipe also being of considerable length to provide a body of liquid of sufficient mass and inertia to ensure the production of adequate pressure on the liquid in the inlet pipe to store said energy and to provide sufficient momentum to exert a pull upon the body of liquid in the inlet pipe, a cylinder in free communication with said system at the junction of said inlet and delivery pipes, a piston adapted to reciprocate in said cylinder and means for reciprocating said piston at a rate of speed unrelated to the dimensions of natural period of vibration of said system but adapted to intermittently apply pressure to the bodies of liquid contained in said pipes, whereby the liquid in the delivery pipe is moved therethrough during the application of pressure so that it tends to separate from the liquid in the inlet pipe, and further whereby the pressure applied to the inlet part of the system resiliently stores energy uniformly throughout the whole volume thereof until such energy is released, whereupon the movement of the liquid in the delivery pipe becomes operative on the liquid in the inlet pipe to transfer a portion thereof to the delivery pipe and thereby permit the inflow of further liquid through the inlet valve, cut-off means provided in the delivery pipe for turning the flow of liquid

on or off as the need requires without stopping the operation of said piston, and normally inoperative and ineffective means in the system for absorbing and nullifying only the pressure impulses due to a continued operation of the piston while the cut-off means is closed.

3. A pumping system for liquids comprising an inlet pipe with a non-return inlet valve at its intake end, a normally open unobstructed delivery pipe having one end in direct communication with the outlet end of the inlet pipe, the inlet pipe being of considerable length to provide sufficient inherent resilience and expansion for the storage of energy resulting from intermittently applied pressure, the delivery pipe also being of considerable length to provide a body of liquid of sufficient mass and inertia to ensure the production of adequate pressure on the liquid in the inlet pipe to store said energy and to provide sufficient momentum to exert a pull upon the body of liquid in the inlet pipe, a cylinder in free communication with said system at the junction of said inlet and delivery pipes, a piston adapted to reciprocate in said cylinder and means for reciprocating said piston at a rate of speed unrelated to the dimensions or natural period of vibration of said system but adapted to intermittently apply pressure to the bodies of liquid contained in said pipes, whereby the liquid in the delivery pipe is moved therethrough during the application of pressure so that it tends to separate from the liquid in the inlet pipe, and further whereby the pressure applied to the inlet part of the system resiliently stores energy uniformly throughout the whole volume thereof until such energy is released, whereupon the movement of the liquid in the delivery pipe becomes operative on the liquid in the inlet pipe to transfer a portion thereof to the delivery pipe and thereby permit the inflow of further liquid through the inlet valve, cut-off means provided in the delivery pipe for turning the flow of liquid on or off as the need requires without stopping the operation of said piston, a chamber in the system adjacent said cylinder and piston, and a spring-pressed piston in said chamber for absorbing and nullifying only the pressure impulses due to the continued operation of the pressure applying piston while said cut-off means is closed, said spring being of sufficient stiffness that in normal operation, when said delivery pipe is open and unobstructed, the spring pressed piston does not move.

4. A pumping system for liquids comprising an inlet pipe with a non-return inlet valve at its intake end, and a normally open unobstructed delivery pipe having one end in direct communication with the outlet end of the inlet pipe, the inlet pipe being of considerable length to provide sufficient inherent resilience and expansion for the storage of energy resulting from intermittently applied pressure, the delivery pipe also being of considerable length to provide a body of liquid of sufficient mass and inertia to ensure the production of adequate pressure on the liquid in the inlet pipe to store said energy and to provide sufficient momentum to exert a pull upon the body of liquid in the inlet pipe, a cylinder in free communication with said system at the junction of said inlet and delivery pipes, a piston adapted to reciprocate in said cylinder and means for reciprocating said piston at a rate of speed unrelated to the dimensions or natural period of vibration of said system but adapted to intermittently apply pressure to the bodies of liquid contained in said pipes, whereby the liquid in the delivery pipe is moved therethrough during the application of pressure so that it tends to separate from the liquid in the inlet pipe, and further whereby the pressure applied to the inlet part of the system resiliently stores energy uniformly throughout the whole volume thereof until such energy is released, whereupon the movement of the liquid in the delivery pipe becomes operative on the liquid in the inlet pipe to transfer a portion thereof to the delivery pipe and thereby permit the inflow of further liquid through the inlet valve, said delivery pipe and said inlet pipe each being adapted to be placed in communication with a body of liquid and provided with branched ends, one of the branches of each of said ends being provided with a non-return valve and the other branch being fitted with a manually controlled valve, rendering the pipes interchangeable in function to effect transfer of liquid in either direction.

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A Novel Form of Refrigerator

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When a gas is allowed to flow out along the radius in a rapidly rotating enclosure, it is compressed and therefore heated. The enclosure may be supplied with a coil of pipe carrying a cooling liquid to remove the heat of compression as formed in the gas. On guiding the gas back to the axis, it is cooled by its own expansion, and a maximum ΔT of about 150°C occurs. The work of expansion is determined by the rotor demand, not by the gas temperature or by the gas volume change. Both warm and cold fluids enter and leave the enclosure axially. Analysis shows that the cooling, ΔT, is given by $\omega^2r^2/2C_p$. The rotational energy of the enclosure is not drawn upon by the steady fluid flows. The cooling liquid moves convectively through its circuit but the gas must enter the enclosure at a pressure somewhat above that at exit. A possible mechanical design is sketched. Maximum peripheral speeds allow 19 cal./gram to be extracted from the gas flow giving for air a drop of 77°C from room temperature. This is more than sufficient for most commercial refrigeration. Use of a favorable gas in the rotor gives sufficient ΔT to be very effective in gas liquefaction. Rotors may have their flows in series to increase the ΔT, and the gas may be at a supercritical pressure to keep it homogenous till released from the pressure. The device is expected to have high efficiency as a refrigerator. By utilizing the rejected heat, as Kelvin suggested, the device may be used for domestic heating.

1. INTRODUCTION

IN the early development of mechanical refrigeration, the compression-cooling-expansion cycle using air as working fluid occupied an important place. But for a variety of reasons it has been displaced by the somewhat similar cycle employing liquid-vapor as working material, especially for relatively small temperature drops while for large temperature drops, as for example in liquefying air, gas is still used for the working material.

In the theoretical discussion accompanying the early experimental work, Kelvin pointed out the dominating advantage, in the heat theory, of an engine applying the Carnot cycle. The practical application is, however, faced with serious difficulty, among which may be noted the complete failure of all attempts to compress a gas isothermally in any reasonable time. The corresponding problem of expanding the gas adiabatically has been only partially met in the uniflow expansion engines. The bulky machines for handling gas at atmospheric pressure magnify the overhead, the friction losses, and the conductive losses. The following scheme meets some of these difficulties in what appears to be a new way.

2. GENERAL THEORY

To study the behavior of the working material separate from the mechanical difficulties, suppose that the bent pipe (Fig. 1) *a b c d e f* of strong nonconducting material, may be rotated in the bearings *B* and *B* at any desired very high speed.

Consider a flow of air adiabatically from *a* to *f*. As the air passes along *bc* out from the axis, it will be subjected to a centrifugal force toward *c* so compressing the air along *bc* and in *cd* to a pressure above that in *ab* and raising its temperature. As this compressed warmer air moves in toward the axis along *de*, it is progressively relieved of the centrifugal force and expands back to its former pressure and temperature by the time it enters *ef*.

Next let us supply the section *bc* with suitable cooling means so that the heat of compression along *bc* will be carried away. This compressed gas may thus approach *d* with approximately the same temperature as at *a* and will be strongly cooled by the adiabatic expansion along *de*. The resulting drop in temperature will depend on the ratio of the pressures at *d* and *e*, and on the character of the particular gas used. Under the most favorable circumstances it appears possible to obtain up to 150°C drop.[*]

It will be observed that the gas along *de*, being colder at corresponding radial points than the

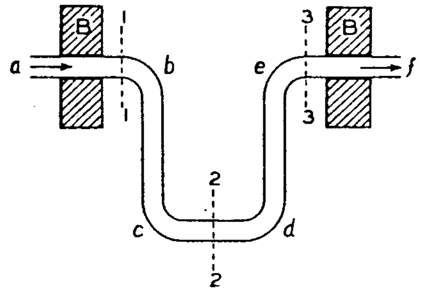

Fig. 1.

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[*] J. W. Beams has observed some such cooling under quite similar circumstances. See *Science in Progress* (Yale University Press, 1940), edited by G. A. Baitsell, p. 246.

gas along *cb*, will be denser also. Thus under flowing conditions the pressure gradient along *de* will be larger than that along *bc* produced by the centrifugal force; and hence for flow to be maintained, the pressure in the gas at entry at *a* must be sufficiently greater than that at *f* to overcome this opposing pressure, and also of course to overcome any frictional effects in the gas flow. Moreover, the external work thus carried into the system may possibly satisfy the requirement of the second law for the expenditure of work to raise the heat from the temperature of the gas being cooled by expansion in the arm *de* to the temperature of the cooler in the arm *bc*.

This mechanical system thus consists of a compressor with concurrent abstraction of compressional heat in *bc*, and an adiabatic expansion engine in *de*. They are very directly coupled mechanically. In a *steady running state*, every gas particle in moving from *b* to *c* acquires a very large kinetic energy due to the large circumferential velocity. Along *cd* this kinetic energy does not change. But as the gas particle moves back toward the axis along *de*, it loses this energy to the tube. Since the number of particles in unit time moving outward across any cross section in *bc* must be exactly equal to the number moving inward across any cross section in *de*, the circumferential velocity energy of the gas makes no major demands on the rotary energy of the tube.

3. INTERCHANGE OF ENERGY BETWEEN PERIPHERAL VELOCITY AND RADIAL WORK

When a small body m at A in Fig. 2 is moved a distance dr inward along the radius r, the work $m\omega^2 r dr$ is done against the centrifugal force. If at the same time the radius is rotating about O, the path followed by m in the resultant motion is along the spiral AC and the displacement ds along the spiral is not normal to the radius r. The centripetal force along the radius now has a component in the direction of the spiral motion, and thus can exchange energy with the moving body m so altering its kinetic energy and its velocity. In the case of interest here the peripheral velocity is very much greater than the radial, so that the angle θ between the spiral and the radius is only slightly below 90°. Thus the work dw done by the component of the radial force, F, over the distance ds along the spiral is, $dw = (F \cos \theta) ds = F dr = m\omega^2 r dr$. Thus the work done in the arm *de* (Fig. 1) by the gas pressure in moving the gas particle m against the centrifugal force, appears as work done in the direction of the spiral and will accelerate the angular velocity of *de* unless the energy is withdrawn elsewhere from the system. Conversely, the gas particle moving out the arm *bc* draws upon the peripheral kinetic energy of the system for (a) its needed increment of peripheral kinetic energy, and (b) the energy needed to supply the work done by the centrifugal force along the radius, which work is spent in compressing the gas.

For the case of interest here the angular velocity, ω, is to be held constant by outside means. As m moves out the arm *bc* from the axis to the end of the radius, r, it acquires a peripheral kinetic energy of $\frac{1}{2} m\omega^2 r^2$ and has work done upon it by the centrifugal force of

$$\int_0^r m\omega^2 r \, dr = \tfrac{1}{2} m\omega^2 r^2,$$

so that m draws upon the energy of the rotating system, or upon its regulating energy store, for a quantity of energy $m\omega^2 r^2$.

Similarly in *de* the particle loses its peripheral kinetic energy, $\frac{1}{2} m\omega^2 r^2$, to the tube as it moves toward e against the force $m\omega^2 r$, against which it does also the work $\frac{1}{2} m\omega^2 r^2$, which work must be drawn from the enthalpy of the gas, as appears in Eq. (2). This loss of enthalpy occurs for purely mechanical reasons independent of whether or not the heat of compression has been abstracted in the arm *bc*. Only in case the kinetic energy (heat) of the particle falls below

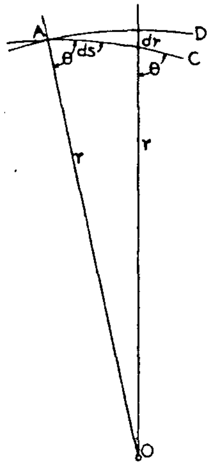

FIG. 2. Spiral path of gas particle.

the $\frac{1}{2} m\omega^2 r^2$ will the demand fail to be met, in which case also the gas will fail to pass through the tube.

It thus appears, as also in the case of the peripheral velocity energy, that the radial work requirements in *bc* and in *ed* are set entirely by the rotation requirements, independent of the means for supplying or absorbing this work. That is, the radial work requirement at every radial point in *bc* is exactly matched at the corresponding radial point in *ed*. Since also the mass flow through every cross section of the tube *af* is the same in the steady flow state, it follows that the radial work requirement in *bc* is exactly equal and opposite to that in *de*, and hence that the radial work in the steady gas flow makes no net demand on the rotary energy of the system.

4. UNIQUE ASPECTS OF THIS EXPANSION ENGINE

The work done by the expanding gas is thus not set by the character of the (p, v, t) relations of the particular gas but by the demands of the rotating system; that is for the element of the gas to reach the axis this quantity of work must be supplied to the rotating system. This is a unique difference from all other types of expansion engines in which the falling of the pressure due to falling gas temperature lessens markedly the work extracted by the piston.

This situation, where a definite force can be exerted on each molecule of the working fluid without the immediate presence of solid bodies to accomplish it, allows of a second unique difference with all other types of compressive-expansive engines. The gas molecules may be made to move among surfaces cooled or warmed by a flow of liquid from the outside without interfering in any way with the work relations in the gas. Sufficient area of such surfaces with an ample flow of liquid will hold the temperature of the compressing or expanding gas as constant as that of the liquid. That is, the compression or expansion may be made closely isothermal. Conversely, the absence of such surfaces in a guide tube of poorly conducting walls allows the steadily flowing gas to bring the wall at each place into temperature equilibrium with the gas at that place, so providing nearly ideal conditions for adiabatic changes.

5. THERMODYNAMICAL RELATIONS

Consider the energy gains to and losses from the volume of the tube between cross sections *1*, *2*, and *3* in Fig. 1. Take account of those due to the gas flow and neglect those due to any heat exchanges through the tube wall. Express all the energy quantities in heat units. For unit mass transfer across *1*, the *gains* to the space between *1* and *2* are: (1) u_a = intrinsic energy coming with the gas in across *1*; (2) $p_a v_a$ = work done by the outside force in driving specific volume v_a in across *1*; (3) $\frac{1}{2}\omega^2 r^2$ = work done by the tube to give peripheral velocity to the gas between *1* and *2*; (4) $\frac{1}{2}\omega^2 r^2$ = work done by the centrifugal force on the unit mass between *1* and *2*. And for unit mass transfer across *2* from the space between *1* and *2*, the *losses* are: (5) u_c = intrinsic energy passing out with the gas across *2*; (6) $p_c v_c$ = work done to force the specific volume v_c out; (7) $\frac{1}{2}\omega^2 r^2$ = kinetic energy (peripheral) carried out with the gas; (8) q = heat carried away by the cooling liquid. It will be noted that (1) offsets (5) since the intrinsic energy of a gas is a very small function of the volume, and the temperature at a and c are alike; that (2) offsets (6) since $p_a v_a = RT = p_c v_c$; (3) offsets (7); and therefore $\frac{1}{2}\omega^2 r^2 = q$. It may be inferred that q is the work done in compressing the gas isothermally in *bc*, so that

$$q = \tfrac{1}{2}\omega^2 r^2 = \int_{v_c}^{v_a} p dv = RT_c \log_e (v_a / v_c)$$
$$= RT_c \log_e (p_c / p_a). \quad (1)$$

That this relation follows from the steady state conditions for the gas in the rotating arm *bc* will be shown below.

Similarly, consider the energy gains and losses to the volume of the tube between *2* and *3*. For unit mass transfer in across *2*, the *gains* are: (1) u_c = intrinsic energy entering with unit mass across *2*; (2) $p_c v_c$ = work done to force the specific volume in; (3) $\frac{1}{2}\omega^2 r^2$ = peripheral kinetic energy entering with the gas; (4) $q = 0$ since there is no heat input to offset the expansion work. And for unit mass transfer out across *3*, the *losses* are: (5) u_e = intrinsic energy passing out with the gas across *3*; (6) $p_e v_e$ = work done to force the specific volume out; (7) $\frac{1}{2}\omega^2 r^2$ = peripheral kinetic energy transferred to the tube between *2* and *3*; (8) $\frac{1}{2}\omega^2 r^2$ = work done by the gas against the radial force between *2* and *3*.

Again it will be noted that (3) offsets (7), and hence on the basis of the conservation of energy,

$$(p_c v_c - p_e v_e) + (u_c - u_e) - \tfrac{1}{2}\omega^2 r^2 = 0,$$

$$R(T_c - T_e) + C_v(T_c - T_e) = \tfrac{1}{2}\omega^2 r^2,$$

and

$$C_p(T_c - T_e) = \tfrac{1}{2}\omega^2 r^2, \quad (2)$$

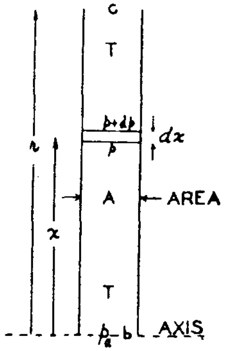

FIG. 3. Steady state in arm bc.

since the general gas law $pv=RT$ holds. Since *enthalpy* is defined as $(u+pv)$ and the first equation may be rewritten as

$$(u_c+p_cv_c)-(u_e+p_ev_e)=\tfrac{1}{2}\omega^2r^2,$$

we note that the change in the enthalpy of the gas between the points c and e is $\frac{1}{2}\omega^2r^2$.

In this situation, T_e, ω, and r are chosen for the experiment, C_p is fixed by the choice of working gas, so that T_e can be calculated from Eq. (2).

From the adiabatic and the general gas laws it is readily shown that

$$T_c/T_e=(p_c/p_e)^{(K-1)/K} \qquad (3)$$

where $K=C_p/C_v$. If the discharge pressure p_e be chosen, p_c may be calculated from Eq. (3). This p_c may then be used in Eq. (1) for isothermal compression to give p_a.

In case the gas used does not obey the simple gas laws sufficiently well, and if the gas has been sufficiently well measured to have available engineering charts, these charts may be readily used to carry out the above calculations with closer correspondence to the behavior of the particular gas.

6. WORK RELATIONS IN THE ARMS bc AND de IN THE STEADY STATE

It will next be shown that Eq. (1) is satisfied when steady state flow conditions have been established in the isothermal arm bc as it is rotated at constant angular velocity ω. Under these established conditions in the arm bc, Fig. 3, consider a layer of small thickness dx fixed at an arbitrary radial position x in bc. The pressure drop dp across dx, and the density ρ of the layer have become constant, although a constant stream of molecules passes through dx. The mass in the layer has also become constant at ρAdx. Since the layer is constrained to rotate with the tube, it is subject to the centrifugal force $(\rho Adx)\omega^2x$. The mass of this layer is in equilibrium when this centrifugal force is just balanced by the difference of the bombarding forces on the two faces, so that

$$A(p+dp)-A(p)=Adp=\omega^2x\rho Adx,$$

$$\omega^2xdx=dp/\rho=RT_a(dp/p). \qquad (4)$$

Since ω and T_a are constants, integration gives

$$\tfrac{1}{2}\omega^2x^2=RT_a\log_e(p_x/p_a), \qquad (5)$$

holding for any point x in the rotating tube. When $x=r$,

$$\tfrac{1}{2}\omega^2r=RT_a\log_e(p_c/p_a). \qquad (1)$$

The right-hand member may be recognized as the work required to compress the gas isothermally from the pressure p_a to p_c which work is converted into the heat of compression q. All of which confirms the conclusions in the discussion of Eq. (1) above.

A similar argument may be based upon Eq. (4) for the *adiabatic* expansion in the arm de, Fig. 1. It can thus be shown that the steady state distribution is satisfied by the conditions of Eqs. (2) and (3).

7. THE GAS CYCLE

The gas cycle involved in this process may be described as in Fig. 4 by a pv diagram which is here drawn to scale in arbitrary units. The condition of the gas at the starting point α is set by its pressure $p_\alpha=p_e$ and its temperature T_a. The pressure p_α is raised to p_β by the auxiliary compressor which process is here assumed as isothermal and $T_a=T_\alpha=T_\beta=T_\gamma$. The pressure $p_\beta=p_a$ is raised to $p_\gamma=p_c$ in the rotor, Fig. 1, bc, under reasonably isothermal conditions. The pressure is dropped in the adiabatic expansion $\gamma\delta$ all the way back to p_α and the temperature T_a drops to T_e. The gas is warmed at constant pressure p_e, from T_e back to T_a, completing the cycle.

The two stages $\alpha\beta$ and $\beta\gamma$ form a continuous curve in Fig. 4, being continuous in pressure and in volume, and really form one step, even though the work for each is supplied separately and the heats of compression are disposed of separately.

The net external work done per cycle on the system is given by the area $\alpha\beta\gamma\delta$, and the position of β on $\alpha\gamma$ is to be so chosen that the compression $\alpha\beta$ supplies this net external work. So that

$$RT_a\log_e\frac{p_\gamma}{p_\alpha}-C_v(T_a-T_e)-p_e(v_\alpha-v_\delta)$$
$$=RT_a\log_e(p_\beta/p_\alpha),$$

$$RT_a\log_e\frac{p_\gamma}{p_\alpha}-RT_a\log_e\frac{p_\beta}{p_\alpha}$$
$$=C_v(T_a-T_e)+R(T_a-T_e),$$

and

$$RT_a\log_e\frac{p_\gamma}{p_\beta}=C_p(T_a-T_e)=\tfrac{1}{2}r^2\omega^2 \qquad (6)$$

which checks with Eqs. (1) and (2). Moreover, inserting the values from line 1, Table I, and solving for p_β checks with the value there obtained from the engineering charts for air.

8. EFFICIENCY

The refrigeration efficiency, E, is, therefore,

$$E=\frac{\text{total heat energy abstracted}}{\text{net external work supplied}}$$
$$=\frac{RT_a\log_e(p_\gamma/p_\beta)}{RT_a\log_e(p_\beta/p_\alpha)}=\frac{\log_e(p_\gamma/p_\beta)}{\log_e(p_\beta/p_\alpha)}. \qquad (7)$$

The refrigeration efficiency of a reversed Carnot cycle may similarly be expressed as

$$E'=\frac{Q_2}{W}=\frac{Q_2}{Q_1-Q_2}=\frac{T_2}{T_1-T_2}. \qquad (8)$$

Before applying this to the rotor it is to be noted that for E', all the heat Q_2 must be lifted from T_2, where it is absorbed, to T_1, where it is discharged. In the rotor, however, while all the gas drops from T_a to T_e, the heat is not all raised from T_e to T_a. If an element of gas, dm, be followed down de, Fig. 1, it will be observed to lose heat steadily and continuously all the way, each loss in enthalpy, dq, being then and there converted into work and handed to the tube de. This work is transmitted by force transmission through the rigid tube to the part bc where it supplies the heat of compression which

TABLE I.

No.	Gas	Molec. Weight	c_p g.cal/g.deg	F.P.	B.P.	q g.cal.	$-\Delta t=t_a-t_e$	t_a	t_e	P_a atmos.	P_c atmos.	P_e atmos.	E	E'
1	Air[1]	28.8	0.248		−190°C	19.1	79.0	20°C	−59°C	1.2	3.4	1	5.71	2.71
2	Air[1]					19.1	81	20°C	−60°C	12.3	30	10		
3	Air[1]					19.1	81	−39°C	−120°C	1.4	4.8	1		
4	Air[1]					19.1	80	−25°C	−105°C	1.4	4.4	1	3.40	2.10
5	Air[1]					19.1	85	−105°C	−190°C	2.8	13	1	1.52	0.98
6	Air[1]					38.2	173	−17°C	−190°C	6.9	52	1		
7	Argon	39.9	0.123	−189.2°C	−185.7°C	19.1	155	−30°C	−185°C	2.75	14	1		
8	Neon	20.2	0.246	24.6°K	27.3°K	12.3	50	77.2°K	27.3°K					
9	Hydrogen[2]	2.0	3.42	14.2°K	20.35°K	19.1	8.6	29°K	20.4°K	1.25	2.5	1		
10	Helium[2]	4.0	1.24	1.3°K	4.21°K	19.1	14.4	34.8°K	20.4°K	1.30	3.4	1		
11	Helium[2]					19.1	17.3	21.5°K	4.2°K	14	62	1		

[1] H. Hausen, *Der Thomson-Joule Effect* (Forschungsarbeiten von verein Deutscher Eng. 1926), Vol. 274.

[2] Keesom and Houthoff, Comm. of Leiden Lab. (Supp. No. 65e, 1928).

[3] Keesom and Houthoff, Comm. of Leiden Lab. (Supp. No. 65d, 1928).

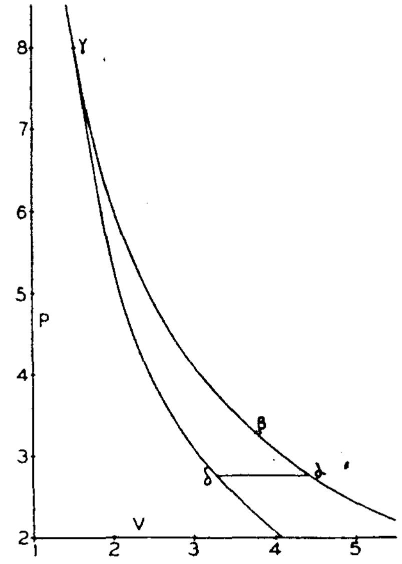

FIG. 4. Gas cycle.

is discharged to the cooling water at T_a. Thus the heat is being lifted at every point and from every temperature along de, to be all discharged at the temperature T_a in bc.

To apply this Carnot refrigeration efficiency E', we have

$$dq = E'dw = [T_2/(T_1 - T_2)]dw,$$

where dw is the work necessary to lift the heat dq from T_2 to T_1. The relation between dq and dt is given by $dq = C_p dt$, hence,

$$dw = [(T_1 - T_2)/T_2]dq = C_p[(T_1 - T_2)/T_2]dT_2,$$

where the integral is to be taken with T_1 fixed at T_a and T_2 varying from $T_1 = T_a$ down to the limit $T_2 = T_e$. Therefore inserting these special variables, we have

$$-w = -\int_{T_a}^{T_e} C_p\left(\frac{T_a}{T} - 1\right)dT$$
$$= C_p T_a \log_e \frac{T_a}{T_e} - C_p(T_a - T_e).$$

In the adiabatic expansion, to which this may now be applied, the states are characterized thus:

$$\left.\begin{array}{lll}\text{Initial state} & T_a p_\gamma & v_\gamma \\ \text{Final state} & T_e p_\alpha & v_\delta\end{array}\right|$$

$$\therefore \quad \frac{T_a}{T_e} = \left(\frac{p_\gamma}{p_\alpha}\right)^{(K-1)/K} = \left(\frac{p_\gamma}{p_\alpha}\right)^{R/C_p},$$

$$\therefore \quad -w = C_p T_a \frac{R}{C_p} \log_e \frac{p_\gamma}{p_\alpha} - C_p(T_a - T_e)$$
$$= RT_a \log_e \frac{p_\gamma}{p_\beta} + RT_a \log_e \frac{p_\beta}{p_\alpha} - C_p(T_a - T_e)$$
$$= RT_a \log_e \frac{p_\beta}{p_\alpha}.$$

Here w is the net external work, supplied to the cycle, calculated as necessary by applying the Carnot refrigeration efficiency to the heat as it is raised to the discharge temperature. This net external work is the same as that which appears in Eq. (7) and so gives the same value of the efficiency.

Hence it is now proved that the external work supplied by the auxiliary compressor in raising the pressure from p_e to p_a supplies the quantity of external work necessary to raise the temperature of the discharged heat to the temperature of the cooling fluid in the rotor, so satisfying the demands of the Second Law.

That the efficiency, E, Eq. (7), obtained in the rotor is decidedly better than E' as defined in Eq. (8), may be readily appreciated from the two sets of circumstances involved. E' corresponds to the cooling of a fluid-flow by a boiling liquid as commonly practiced. In comparison with this, it is to be noted that the cooling in an ideal exchanger, as in $b\alpha$ of Fig. 4, is assumed to have unit efficiency. The values of E and of E' for some cases are given in Table I, where the E's are pure numbers, that is, the heat discharged and the work expanded are both expressed in the same energy unit. The advantage of E over E' is very material.

9. REVERSIBILITY

This rotor process as described above is close to thermodynamically reversible. If, with all the conditions of pressure, temperature, and flow as described, the pressure supplied at a, Fig. 1, is alone decreased a little, the flow of gas will reverse going now from f toward a, compressing adiabatically from e to d, and expanding isothermally with absorption of heat from c to b, and delivering the gas at a at a pressure raised above that of f, so converting some heat into work. If now with suitable reservoirs of gas at a and at f to maintain this reversed flow, the fluid entry temperature at a be raised a little, the flow will be accelerated, so that the pressure previously at a may be restored, and the flow remain reversed. Thus by manipulation of the pressure (work) and temperature (heat) variables, the process may be run in either direction with, it would appear, relatively small changes.

This ready reversal of the process means that its "intrinsically irreversible" steps are relatively unimportant, that specifically the heat transfer by conduction takes place with small temperature drops and that the small friction allows the gas flow to take place with small pressure drops. That, in fact, the rotor process is not far from "thermodynamically reversible" and hence correspondingly efficient.

10. COOLING LIQUID CONDITIONS. CONVECTION

The cooling liquid entering the rotor near the axis is also compressed as it moves toward the periphery and expanded as it returns toward the axis. Since liquids are normally much denser than gases, the liquid used rises in pressure toward the periphery much more than the gas. The liquid should on this account be contained in very strong walled tubes of rather small diameter wound in cylindrical layers, with the gas moving among the tubes. Liquids also change their temperature on compression, rising in temperature if the coefficient of volume expansion is positive. These temperature changes are very much smaller than those for gases—a fact which makes this rotor device feasible. Thus, for the change of about 400 atmos. to be expected at the limiting linear peripheral velocity, water initially at 20° rises less than 1°C, and organic liquids 2°–7°C. Water is unlikely to be used in the range of negative expansion coefficient since the convective force will oppose the cooling flow and, even though it is apparently possible, to achieve a small total drop in temperature for the pressure change of 400 atmos. Gases differ considerably in their temperature changes on the same adiabatic compression, but these differences are difficult to employ for such cooling purposes as are needed here.

For the liquid to cool the gas most effectively, the two flows should probably run countercurrent. Hence, the liquid entering on the axis should proceed directly to the periphery out of contact with the gas and return to the axis under the best possible contact with the compressing gas. If the heat absorbed raises the temperature and expands the liquid, the column of liquid returning to the axis becomes less dense than that going to the periphery. The two radial columns become unbalanced in the large artificial gravity field, producing a very considerable pressure difference which drives the liquid around its circuit. The work necessary for this drive, as in ordinary convective flows, is drawn from the heat energy causing the expansion. This makes unnecessary any other means for circulating the liquid and at the same time does not draw on the rotary energy. As the heat to be absorbed is increased, the tendency for the liquid temperature to rise will be largely offset by the increase in driving pressure difference increasing the mass flow of the liquid. This flow is thus quite self-regulating, changing roughly proportionally to the mass flow of gas through the rotor.

11. GAS CONVECTION

A somewhat similar convective situation is present in the gas flow. If a small volume of gas loses heat by contact with the tubes carrying the liquid, its density increases, throwing it out of equilibrium with its surrounding gas. This cooled gas is accelerated by the large artificial gravity out the radius and hence up the pressure gradient. This movement ceases when the compressional heating brings this cooled gas back into equilibrium with its new surroundings. Such movements setting up heavy convection in the gas greatly facilitate the transfer of heat between the gas and the liquid. If any radial channel in the gas has no effective means of losing its heat of compression while parallel neighboring channels have, the gas in the former channel will lie effectively quiescent while the movement takes place in the cooled channels. Hence in the design there is no need to be extremely particular to avoid any such channels, since very little flow will take place there.

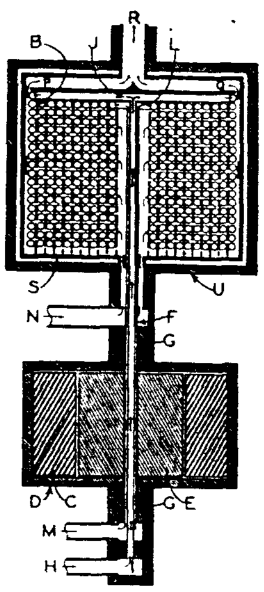

FIG. 5.

12. REFRIGERATION PROCESS IS INDEPENDENT OF ROTARY ENERGY OF ROTOR

From the discussions above, it would appear that after a steady state has been reached in the operation of the rotor, the rotor drive will not be called upon to supply any major quantities of energy on account of the flows. Energy for driving the gas and liquid flows are separately supplied, the gas from the raised inlet pressure and the liquid from the heat absorbed. Rotary energy is lost in bearing friction and in gas friction on the outside of the rotor, but these should not be materially affected by the gas and liquid flows. Neither the gas nor the liquid can leave the rotor exactly on the axis, so that in each a small escape of rotational energy in the form of fluid turbulence will occur. Except in such minor ways, the refrigeration process does not draw upon the rotational energy of the rotor.

13. DESIGN OF ROTOR UNIT

The considerations detailed above, when combined with ordinary mechanical considerations, lead to a design for the rotating vessel as represented diagrammatically in Fig. 5.

The cylindrical case *S* is made of the strongest available metal, e.g., chrome nickel steel, with a wall thickness sufficient that the stress from the internal gas pressure will not add very materially to the unit stress. The stress limiting the angular speed will occur in this wall or in the outer layer of tubing carrying the liquid. This cylinder *S* rotates about its axis with moderate clearances in a heavy steel cylinder *U* which will serve as a guard against flying parts in case of trouble and also permit *S* to rotate in a partial vacuum. *S* is carried on the entrance and exit tubes, a convenient one of which may be used to drive it.

Since for any effective use the speeds required are high, it will be desirable that the driving power have a very fixed upper speed limit such as a synchronous motor or any induction motor driven by high frequency electrical power. Since, as indicated above in the theory, the power demand from this rotor by the process is small, the size of this motor will be set partly by the time required to bring the rotor up to its working speed.

C and *E* of Fig. 5 are intended to illustrate a high frequency induction motor. The field magnets of the stator *C* are mounted inside the steel case *D*, and the squirrel cage rotor *E* is carried directly on the cooling-fluid exit pipe *F* running in the bearings *G* and *G*.

The cooling fluid—water, alcohol, or other light mobile fluid—enters the apparatus at *H*, the rotating inlet tube at *I*, and divides at the manifold *J* for the represented case into 8 strands. These proceed to the periphery directly where they are wound as 8 parallel strands in a single layer on the perforated steel cylinder *B*. These perforations are narrow slots running circumferentially for rather short angular distances.

The 8 tubes coming from the manifold *J* and wound parallel on the cylinder *B* pass over the lower end edge of this cylinder to be wound back to the upper end on the next inner cylinder. These tubes are so wound and the cylinders shrunk on them as to make a very rigid strong assembly. The tubes are suitably spaced to allow effective gas flow among them and are provided several times per circumference with septa parallel to the axis and to the radius so that the gas will always move around with the same angular speed as the rotor. These septa are formed by inserting between the tubes during winding pieces of metal shaped to fit the spaces between adjacent tubes of one layer and between successive cylinders. The tubes finally reach the manifold at *L* and join the liquid outflow tube, to leave the rotor at *M*.

The compressed gas enters at *N*, passes along the axis, spreads out among the tubes, gathers along the periphery, to pass over the edge of the disk *PQ*, to move in to the exit at *R* on the axis. The space between *P* and *Q* has a number of radial-axial septa to assure that the gas has always the same angular velocity as the rotor.

It is to be noted that the tube layers at successively greater coil diameters run at successively smaller angles to the plane normal to the axis. Also that the slots cut in the cylinders between these tube layers should for maximum strength be cut parallel to this plane. Hence it is not possible by any choice of plane of section to present a clear picture of the gas flow pattern among these tubes.

It will, however, be evident that the gas flows parallel to the pipes in each layer for small angular distances between steps from one layer to the next. If these angular distances be kept alike, the linear distances involved increase with the radius. Since the work done in compressing the gas along the radius increases proportionally with the radius, the heat to be transferred per unit area of tube remains the same. Since the gas density increases with radius, the gas velocity past the adjacent metal parts decreases in successive layers toward the periphery. Moreover the huge artificial gravity field greatly exaggerates convective movement in the gas. The cooling surfaces on the tubes are thus swept by movement of the surface gas layer, and the heat exchange is greatly accelerated. The mass contact of the gas and its convective momement combine to make the heat exchange very effective.

14. ROTATIONAL SPEED LIMIT

The highest peripheral linear speed at which a machine like this may be run without failure sets the limit on the energy extraction from unit mass of gas. As long as this peripheral linear speed is kept unchanged, the diameter and length of the rotor may be determined by the desired gas flow. Analysis also shows easily that for constant peripheral linear speed the effect of the radial size of the rotor on its strength may be disregarded in a good first approximation. Stodola[1] in his treatise on turbines lays down this limit very emphatically at 1300 ft./sec. (4×10^4 cm/sec.) for the highest tensile strength steels. Beams[2] working with high speed centrifuges says that 8×10^4 cm/sec. is possible. Stodola's limit which may reasonably be used here leads to an energy extraction of 19.1 cal./gram. Since this quantity goes up with the square of the speed, Beam's limit, which experience may show feasible, gives 76 cal./gram.

15. CONTROL OF WORKING RANGE OF TEMPERATURE

It is evident from the above that the temperature range over which the rotor will lift heat depends first upon the gas chosen (the smaller its specific heat, the larger the step) and second upon the linear peripheral velocity given the rotor. Following the choice of working gas, the peripheral velocity can be adjusted to give the desired temperature step. Unfortunately the refrigerating capacity falls in the same proportion as the temperature step when the peripheral speed is reduced.

It is possible, however, to maintain the capacity of the machine but to fix the drop in temperature economically at less than its full drop. The part of the rotor, *PQ* as described in Fig. 5, in which the expansion to the axis takes place, will be increased in axial length to receive a coil like that in the compressional part. The outer diameter of this coil will however coincide with the place in the adiabatic expansion where the gas temperature becomes equal to the outlet temperature desired. A liquid to be cooled is circulated through the pipes of this coil by an outside pump as the convective situation opposes the flow. The heat so supplied to the expanding gas, continues to be drawn on for the further isothermal work, but the gas temperature is not allowed to fall materially further. The work required to carry the gas back to the axis is unaffected, and the resulting quantity of heat extracted is also unaffected. But because of the higher average temperature along this expanding column, the average density remains less than when the full adiabatic expansion takes place. Hence the pressure required to drive the gas through the rotor is lessened. This is in agreement with the Second Law in requiring less external work to lift the heat up a smaller temperature step. It should also be noted that the gas escaping from the rotor is below its inlet temperature and is available also for cooling.

16. USE OF COLD GAS TO PRODUCE LIQUID

The normal product of the rotor is a homogeneous gas, *A*, cooled materially below its rotor entry temperature. If *A*'s low temperature lies a few degrees above or below the normal boiling temperature of a second gas, *B*, then pass the first gas *A* through an exchanger counter-current to a flow of *B* compressed well above its critical temperature. The compressed *B* cools as a homogeneous fluid of nearly constant but raised specific heat which after expansion at a valve will be almost completely liquid. The flows of *A* and *B*, for economical running, will need to be so adjusted that *B* in dropping to the entry temperature of *A* will give up just enough heat to raise *A* to the entry temperature of *B*. If *B* can be cooled below its normal boiling temperature enough that the usual liquid Joule-Thomson warming just brings it to its boiling temperature, the whole of *B* will become liquid. In case it is difficult to cool *B* quite that much, it will cool itself by boiling, and its cold vapor may be used to cool more compressed *B*. The work cost of compressing *B*, if done isothermally and reversibly, counts entirely as part of the necessary work cost of liquefying *B*. This is effected (a) by lessening the enthalpy which must be extracted to reduce *B* to a liquid and (b) by raising the temperature at which this enthalpy may be taken from *B*. Incidentally, the compression has spread the heat of condensation over a wide and raised temperature range where it appears through an increase in the specific heat.

17. WORKING GASES

In Table I are listed a group of gases of interest for use in this rotor. Engineering charts based on their measured physical properties are available for three of them. To use these charts recall the equation two lines above Eq. (2), namely

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[1] Stodola, *Steam and Gas Turbines* (McGraw-Hill Book Company, Inc., New York, 1927), p. 240. Translated by L. C. Lowenstien.

[2] J. W. Beams, Rev. Mod. Phys. **10**, 260 (1938).

$$(p_c v_c - p_e v_e) + (u_c - u_e) = \tfrac{1}{2}\omega^2 r^2$$

or

$$h_c - h_e = \Delta h = \tfrac{1}{2}\omega^2 r^2 \tag{9}$$

where $h = u + pv$ = enthalpy.

Consider the rotor process in the reverse direction to which it actually is used. The adiabatic expansion is isentropic being reversible and exchanging no entropy with the outside. Choose the delivery pressure p_e and guess t_e or calculate it approximately as in Section 5 on the assumption of the gas laws. Locate $p_e t_e$ on the chart and move along a line of constant entropy a distance given by Δh of Eq. (9). If the point reached does not leave the chosen t_c, shift the assumed t_e until it does. Thus read p_c, t_e, and S_c.

From this point, $t_c p_c S_c$, consider an isothermal expansion where the S increases by an amount h/T_c. This gives p_a. The numbers in the right hand part of Table I were obtained by these methods except for argon and neon for which usable charts were not available. The subscripts refer to Fig. 1.

The cooling of 79° in line 1 is more than sufficient for most commercial uses in the storing and handling of food, whether frozen or not. The power savings should be very material for the large installations, particularly for the manufacture of ice. The refrigeration capacity of a rotor could probably be increased materially by working from a base pressure P_e of 10 atmos. as computed for line 2 of the table, instead of a P_e of 1 atmos. as in line 1.

18. AIR LIQUEFACTION

For the writer, the greatest interest in the use of the rotor comes in the field of deep refrigeration, for example, as in the liquefaction of gases, particularly of air. Most air liquefaction processes used commercially employ precooling, as for, example, with the usual liquid-vapor carbon dioxide cycle, yielding down to approximately −50°C. In line 3 of Table I air at −39°C will suffer about the same 79°C drop by passing through the rotor. The inlet pressure p_a steps up a little due in part to the decreased specific volume of air at −39°C and in part to the decrease in the thermodynamic efficiency of the refrigeration cycle at the lower temperature. The Hylandt process uses a considerable fraction of its air supply after compression to 200 atmos. to provide a precooling to −120°C for the remainder, the power cost of which is much greater than for that outlined above.

A more effective arrangement, however, is that covered by lines 4 and 5 using two rotors in series following a carbon dioxide precooler. The carbon dioxide cools the inlet air and cooling liquid to −25° for the first rotor. Its outlet air at −105°C is used to cool the entry air and cooling liquid for the second rotor, whose outlet cold air is close to the normal boiling temperature of air.

If experience should show that a linear peripheral velocity of $\sqrt{2}\times 4\times 10^4$ cm/sec. is feasible, the energy extraction of the rotor would be doubled as assumed in line 6. Under these circumstances a single step rotor using air and following the CO₂ cycle cooling would give outlet air close to its normal boiling temperature.

When argon is used as the working material its very small c_p leads to the very large temperature drop of 155°. Argon boils at −185.7°C under 1 atmosphere. Any liquid formed in the rotor will accumulate around the periphery where the extra load may burst the rotor. Still, the argon could be cooled safely near enough to its normal boiling temperature that the compressed air cooled in the exchanger would, on escape through the valve, be 90 percent or so liquid. The vapor part should be used for cooling more compressed air.

19. HIGHLY COMPRESSED GAS IN THE ROTOR

At first glance it looks very advantageous to use in the rotor a gas compressed to well above its critical pressure. The temperature of a gas under such a pressure may be lowered to near freezing conditions without any possibility of separation into two phases, that is, the fluid remains homogeneous. The compressed gas, if cooled slightly below its normal boiling temperature, will be completely liquid on escaping through a valve to atmospheric pressure. Thus the interchanger considered above might be completely avoided and its involved losses not enter the picture. The decreased specific volume of the gas increases greatly the possible mass flow, so increasing the mass capacity of the machine and therefore its efficiency.

On the other hand, there is to be considered the very serious difficulty of transferring a gas under such pressure in and out of the rotor. If the constriction through which the cold liquefied gas expands is placed at the exit from the rotor, and if the gas inlet to the rotor is moved to the central axial position, the necessary gas-tight moving contacts may be reduced to one. The compressed gas may be passed through several rotors in series on a common shaft, each rotor being supplied between rotors with its necessary separately-cooled liquid. The series arrangement may also be used to cope with the decrease in the temperature drop per rotor owing to the increase in C_p by the compression.

At some point in its cooling, the compressed gas approaches the liquid state—that is, it becomes practically incompressible and changes its temperature relatively little during compression. This takes place markedly only below the critical temperature, but from then on the further cooling in the rotor is likely to be ineffective.

The high pressure will necessitate keeping both fluid and gas in small tubes, preferably one inside the other, and wound as the liquid carrying tubes in Fig. 5, but separating the two tubes at the periphery and returning them to the axis separately. The pressure in the tubes increases the longitudinal tension somewhat but not enough to reduce the maximum rotational speed very materially. It would seem that the possible usefulness of this plan using highly compressed gas in the rotor can only be determined by considerable experimental work.

20. HYDROGEN AND HELIUM LIQUEFACTION

It is interesting to inquire whether such a rotor could serve to reach the very low temperatures of liquid hydrogen and of liquid helium. Lines 8–11 of Table I give some pertinent date on the three possible gases.

It should be noted that the temperatures for this group of gases are given on the Kelvin scale. Neon, line 8, has a low enough c_p (0.246) that it does not need the highest speed of the rotor if used from the boiling temperature of nitrogen (77.2°K) and with nitrogen as the rotor cooling liquid, to escape from the rotor at close to its own boiling temperature 27.3°K. This is of course assuming availability of the very scarce neon.

With liquid neon as cooling liquid and either hydrogen, line 9, or helium, line 10, as working gas, starting at this 27.3°K, the exit gas can be at the boiling temperature of hydrogen (20.4°K).

Starting at this temperature, 20.4°K, with liquid hydrogen as the cooling liquid, helium can escape from the rotor at close to its boiling temperature, 4.21°K.

Too little seems to be known about these liquids to feel safe that none of them will freeze in the rotor at the temperatures and high pressures which well exist there.

In the absence of such misfortunes it thus seems possible to go in a few steps close to the lowest known temperatures.

21. GAS JETS ON THE ROTOR

It may be noted that cooling could also be obtained by discharging the compressed gas at the rotor periphery through jets arranged symmetrically around the rotor and aimed tangentially in the opposite direction to the peripheral velocity, so driving the rotor and cooling the gas. Such jets do not have 100 percent efficiency, and the expansion to the axis appears much to be preferred.

22. ROTOR USE FOR HOUSE HEATING

Kelvin suggested many years ago that the heat delivered by a refrigerating machine be employed for the heating of buildings. When coal is burned under the best conditions in a modern power plant, the work so obtained if applied in an effective refrigerating machine, will raise over the small temperature step needed for this heating, a much greater quantity of heat than that coming directly from the burning of the coal. Since this rotor cooler appears to be the most efficient refrigerator available, it should be effective for the purpose Kelvin suggests.

For the purpose of heat delivery, the rotor-heater should be arranged somewhat differently from the rotor-cooler. The first part of the gas compression should take place adiabatically until the gas temperature rises to that desired for the heating, and in the second part of the compression the gas should flow among the pipes carrying the liquid to be heated, so this part of the compression would be effectively isothermal. The cold discharge gas from the expansion to the axis needs to be warmed by exposure to the building's environment temperature to complete the cycle. Adjustment of the mass flow of the gas and of the speed of the rotor will serve to control the heating.

I wish to express my thanks to Dr. H. Osterberg, Dr. E. E. Miller, Dr. T. A. Murrell, and Dorr Ralph, whose enthusiastic interest in this problem and active discussion of most phases of it have been of great assistance in working out the details of this solution.

"FUELLESS FURNACE" USES FRICTION TO HEAT AVERAGE SIZE HOME FOR "50¢ A DAY"

Eugene Frennette pours hydraulic oil into his prototype "fuelless furnace". The oil, combined with the spinning action of two cylinders, supposedly creates friction which, in turn, produces the heat. "Defies a basic law of physics — a complete hoax," say skeptics. Prototype shown below has been used to provide supplemental heat in Frennette's 12-room house.

FARM SHOW 8500-210TH ST.
JOHNSON BLDG. LAKEVILLE MN
Vol 2 #5 '78

Photo courtesy Montachusett Review

How about this — a fuelless furnace that uses friction instead of fuel to heat an average size home "for only $15 to $16 a month". What's more, it reportedly will sell for less than half the cost of a conventional oil or gas furnace.

Sound too good to be true?

"You bet," say some observers who claim the whole thing's a hoax — that it defies a basic law of physics. But others, including a host of small manufacturers and distributors have jumped at the chance to get in on the ground floor of a "breakthrough" development they feel can help solve the energy crisis. They have invested in franchises and hope to be taking orders for Eugene Frenette's fuelless furnace early next year.

It all started during the winter of 1977-78. It was costing Frenette, father of 12 children — 10 of whom are still at home — a whopping $230 a month to buy fuel oil to heat his huge, old uninsulated 12 room "Pillsbury mansion" in Londonderry, New Hampshire. He launched a crash program to perfect his invention — a simple but unorthodox 'fuelless' furnace which he maintains will be able to heat an average size home for only 50 cents a day and which he feels can be retailed "for $600 to $800."

Frenette installed his prototype friction heater in a 10-year-old washing machine. It's made up of two cylinders spinning in opposite directions. There is a clearance of ⅛ in. between the two cylinders which are lubricated by a quart of light motor oil. Spinning action of the cylinders and resulting friction produces the heat, according to Frenette.

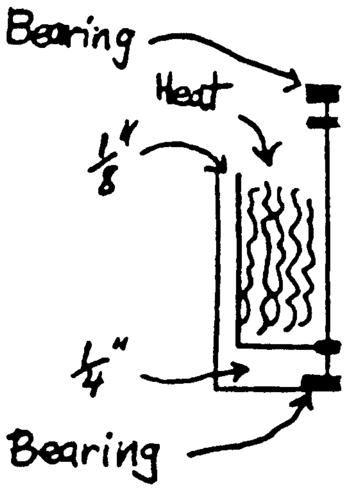
He claims franchised models will be odorless (they don't require any chimney since no fuel is burned and there is no flame, soot or odor) and are as quiet as a refrigerator. All models will plug into a regular 110 volt outlet and will occupy no more space than a washing machine or dryer. Estimated operating cost to heat an average size well-insulated home with a 200,000 btu friction "centric" heater is right at $15 a month (for electricity to operate the motor).

One of the first successful prototypes was built in August by Max Johnston, owner of Johnston's Metal Specialties, Creston, Iowa. "I'll admit I was skeptical at first. Sounded like a hoax to me," says Max, who was hired by the owner of the "Frenette Furnace" franchises for Alaska and Kentucky to build a prototype. Following basic design specs supplied by Frenette, Johnston built a prototype which, in his words, "made a believer out of a lot of skeptics around here, including me. It cost about $800 to build, including about 40 hours of labor. Now that we've built one, we could build another in a lot less time. We estimated its output at between 100,000 and 150,000 btu's. The friction stove produced no odor, made no more noise than you'd get with a furnace motor, and we had no vibration or other problems with the rotating circular drums which create the friction heat," Max told FARM SHOW.

According to Larry Nickerson, Frenette's son-in-law, all franchises except Washington, D.C., and Hawaii, have been sold. Some individuals bought up 3 or 4 states. Cost of a state franchise, based on population, was $2,500 cash, plus an additional down payment payable on availability of the first approved stoves, and a remaining balance spread out over 20 years. The Iowa franchise, for example, was priced at $145,000. Of that, $2,500 was payable immediately to hold the franchise, with $36,250 payable upon availability of Frennette-approved stoves for sale. The balance ($108,750), plus interest is payable over 20 years in monthly installments.

"I bought two states and others from this area bought up many of the other state franchises during the short time they were available," Harold Schweiss, of Sherburn, Minn., told FARM SHOW. Schweiss has hired a firm to produce a working model which was completed and ready for testing just as this issue went to press.

"Frenette came up with the idea but doesn't have manufacturing or marketing expertise," explains Schweiss. "Individual franchise holders are taking the patented idea to local manufacturers to get a working model. These models, subject to Frenette's approval, will then be produced and sold when they've met the usual battery of tests. Eventually, the best features of these prototypes will be combined into production models which will be essentially the same but produced by a number of different manufacturers," Schweiss explains.



Latest 'news' (1-85):
Frenette has 'disappeared'...
Rex Res. associates have successfully
replicated & tested small unit: 300% efficient!

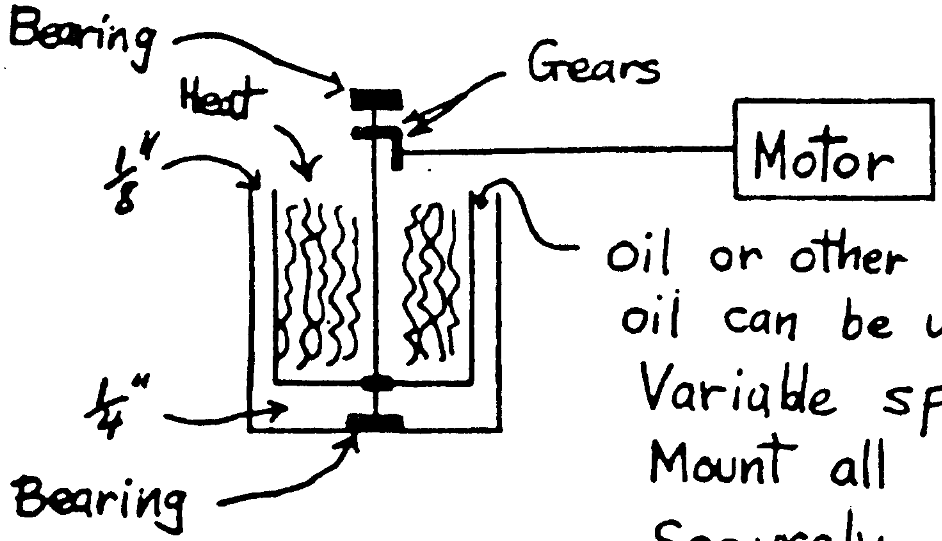
2. Rotary Mechanical Heating Device.

More than one of these units have been built. There are several ways to build a simple working model. The rotor and stationary parts have the same geometry. This may be a cone, cylinder, or cup shape. Start with a small model using two tin cans. The one that fits inside the other should have about a 1/8" clearance around the sides and 1/4" between the inside of both bottoms. Use a variable speed motor or pulley to turn the inside can. One small unit was built that produced about 110° F in a 60° F room. A high temperature oil or other liquid is poured into the outer can. The diameter of the outer can was not more than 3 or 4 inches.

Another unit was built with about a 30 inch diameter cup shape rotor. It used about a 1/2 or 3/4 horsepower motor. The cup does not have to rotate at a high speed. The temperature can easily reach 200° F or 300° F. This device heated a 16 room mansion for around $30.00 per month. It was made into a furnace type unit. The cost reduced by about a factor of ten.

The liquid molecules are set into greater motion due to the rolling friction between the stationary and rotating cups. At the outermost diameter the molecular motion is the greatest and slowly decreases toward the bottom. A good steel or other materi should be used to conduct the heat. A mechanical resonance can also occur in the liquid at the right thermal vibration. This will intensify the heating action. Other constructions will produce cavitation and shock waves. The geometry has a lot to do with its overall performance. Keep it simple in design. Operate at very slow speed.

A sample drawing is included for a basic model



oil or other liquid. a high temperato
oil can be used.
Variable speed motor can be u
Mount all stationary Componen
Securely.

[illegible] NEWSLETTER
1901 E. FIFTH ST. CHARLOTTE NC 28205

FRENETTE

REPRINTED FROM OUR LAST ISSUE
BECAUSE OF PENDING DEVELOPMENTS
-- See late NEWS, next page

B C Communications Inc.

PENNY STOCK COMPANY, BACKED BY "HITK", ACHIEVES OUTPUT vs. INPUT OF 3 x 1+! "OVER-UNITY!"

KINETIC SYSTEMS, INC., 74% owned subsidiary of B C COMMUNICATIONS, INC., under 10¢ per share in the "pink sheets", has achieved outputs of heat and energy equivalents in testing at various laboratories and its own testing at the assembly plant near Atlanta of 300% to 400% of input.

The simplest description of the patented technology is that it embodies a closed chamber, forced hot air, liquid heating system (i.e. a heating system in which a rotor is rotated within a body of liquid within a chamber so as to heat the liquid by friction and with the liquid thereafter being conveyed to an external heat exchanger and then returned to the liquid heater through a circulation system).

However, the system's ability to generate heat is merely an indication that it can produce energy and power of almost any conventional type. Therefore, the technology could be adaptable to produce pure heating, electricity, steam pressure, refrigeration and vehicular motion. The inherent disposition of the technology results not only in a dramatic amplification of energy derived, but also in products which are non-polluting, non-explosive and relatively portable.

NUMEROUS PATENTS GRANTED BY U.S. - PENDING MANY COUNTRIES - ENERGY AMPLIFICATION!

The United States has granted many patents, which are also pending in major countries, with further applications expected to be granted. Present patent numbers include: 4,424,797; 4,483,277; 4,501,231 and 4,651,681 (the last on March 24, 1987).

The patents make no claim to observable energy amplification. Such a claim is not needed to substantiate the patentability of the technology. However, this feature comprises the breakthrough element of the technology and promises to revolutionize many industries for the forseeable future.

While undergoing tests pursuant to nationally recognized test procedures set forth by the American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE) an independent testing lab based in Atlanta has determined that the company's prototypes generate a Coefficient of Performance ("COP") consistently in excess of three to one. The "COP" measures the heat or energy output divided by the total energy input.

The significance of a minimum "COP" of three to one is simply that the technology as configured in the working prototype generated more than three times the energy used to run the machine. This contradicts certain previously taught physics folk lore. The testing laboratory was astounded, to say the least.

Certain preliminary tests on the heating system were commissioned by inventor Eugene Perkins in Sept.-Oct. of 1983 and were conducted by a firm known as Air Techniques, Inc. of Marietta, Georgia. The firm concluded that such heating system does "apparently produce more energy than it consumes, this statement being valid even after allowing for the possibility of a significant margin of error...". They concluded the testing results warranted continued development research testing of such system.

The explanation of the results obtained, as yet unproven, is that the particular configuration of the rotating cylindrical members causes a molecular structure realignment in the liquid medium due to the type of shear forces imparted. These forces are amplified due to the phenomenon of Bernoulli's principle of fluid motion taking place between the rotating and stationary parts.

Bernoulli's principle is the same phenomenon which describes the amplified lift over an airplane wing by fluid air rushing over the top of the wing. As a consequence of the molecular realignment, internal energy is generated within the fluid medium (water in the case of the prototype heater) by virtue of the relatively minor amount of external energy used to operate the mechanism's pump and fan. This can be conceptually viewed as inducing a non-atomic internal reaction which generates excess heat or energy.

When operated as a residential space heater, the water temperature can reach the operating level of 300 degrees F (superheated) in as little as ninety seconds from an ambient room temperature. When operating in a conventional duty cycle mode, the heating chamber is designed to yield an effective COP of eight to one. What are the prospective savings? An electric monthly heating bill of $400 in suburban New York should be reduced to as little as $60 or less. Besides heating unlimited other applications beckon.

... CONTINUED ... B C COMMUNICATIONS subsidiary, KINETIC SYSTEMS, achieves "OVER-UNITY"

HOT WATER ... The production of hot water heating using KSI technology is expected to be a natural application. All that is required is a direct connection from the heating chamber to the water heating element.

STEAM HEATING AND POWER ... The internal function produces superheated water which, in turn, can be used to create steam.

AIR CONDITIONING AND REFRIGERATION ... In this application, a medium containing a very low boiling point such as freon can be employed in an adiabatic cycle along conventional means. However, the compressor unit, when employing the KSI technology is expected to operate much more efficiently when so powered.

GENERATION OF ELECTRICITY ... With a COP of greater than three to one, electrical power may be "multiplied" by adapting the heat generated in the KSI device to operate a turbine or similar device to generate rotational motion which can drive a stator, rotor or armature.

VEHICULAR TRANSPORTATION ... The same principles which could be applied to rotate a turbine may be applied to rotate a crankshaft or propeller. In the case of an automobile for example, the KSI technology could use battery power to drive the pump and fan. A liquid medium with a very low melting point could be used to operate in cold climates. The battery drain would be minor. Rechargeable batteries could be used or recharged by by device configuration. Self-contained and self-sufficient continuously running power systems can be designed to run trucks, trains, ships and planes, eliminating the use of fossil fuel and attendant environmental and political pollution.

MANAGEMENT, FINANCES AND OUTSTANDING SHARES: Company officers are from HITK CORP. and related companies, "HITK" is on National OTC list, with 24 million assets as of fiscal end at 2/28/87, earning $1.29 per share. B C COMMUNICATIONS, INC. has outstanding warrants that could yield over $300,000 to implement licensing/royalty business plan, which projects possible earnings of 250 million annually at end of fifth year.

B C COMMUNICATIONS owns about 74% of KINETIC SYSTEMS, INC., balance private. As of 3/31/87 B C had cash of $373M, total assets of $511M, liabilities of $78M and stockholder equity of $413M. Outstanding shares were 337,123,200, maximum public float, assuming all warrants are converted, will be 75 million shares. Many market makers. TRANSTRAK, INC., pink sheets, owns 30 million restricted B C shares, received for its interest in KINETIC SYSTEMS, INC., giving TRANSTRAK about 9% of B C COMMUNICATIONS, INC

Company address: 777 Summer St., Stamford, CT 06901. Contact: Maureen Dailey at (203) 323-7773. Brokers very familiar with company are at Mary Martin Office of MARSHALL DAVIS, INC., 110 Providence Road, Charlotte, N.C. 28207. Phone 1/800/225-6935, in NC at (704) 338-9661. Mary Martin, supervising principal of this MARSHALL DAVIS INC. office, attended the International Congress of Gravity Field Energy (report on request) and has observed both the KINETIC and TRANSTRAK devices, has met and is in touch with both the inventors/designers of both the KINETIC and TRANSTRAK devices for late news... /JVH

Late news about B.C. Communications (Kinetic)

The "kinetic" power use / power generating system was successfully shown in Europe.

Present at the week-long showing were: • various industrial company represenatives, representatives of two universities, a representative of the German government and a leading European physicist, plus others.

Since returning from Europe the Kinetic Systems group have been in communication with the European group regarding licensing of the devices for manufacture in Europe ... first potentials appear to be space and water heaters.

B C COMMUNICATIONS "KINETIC" division has been engaged in a series of tests in noted United States laboratories. Results expected to be announced by the end of September, if not before. All tests highly satisfactory thus far.

Various negotiations are said to be in progress in the United States with very large companies. While I have not been able to learn the names of these companies it is believed they are among the FORTUNE 500 group. JVH/

United States Patent [19]

Frenette

[11] 4,143,639
[45] Mar. 13, 1979

[54] **FRICTION HEAT SPACE HEATER**

[76] Inventor: **Eugene J. Frenette**, Box 255, Derry, N.H. 03038

[21] Appl. No.: **826,557**

[22] Filed: **Aug. 22, 1977**

[51] **Int. Cl.**^{2} **F24C 9/00**
[52] **U.S. Cl.** 126/247; 122/26
[58] **Field of Search** 126/247; 122/26

[56] **References Cited**

U.S. PATENT DOCUMENTS

2,251,344	8/1941	Tesch	126/247
2,467,425	4/1949	Gobb	236/10

FOREIGN PATENT DOCUMENTS

760213	2/1934	France	122/26

OTHER PUBLICATIONS

A.P.C. Application of Beldimano, Ser. No. 326,790, published May 11, 1943.

Primary Examiner—William E. Wayner
Assistant Examiner—L. I. Schwartz
Attorney, Agent, or Firm—Pearson & Pearson

[57] **ABSTRACT**

A furnace or space heater is operable at low cost by a small electric motor which rotates an elongated cylindrical drum on a vertical axis, within an elongated cylindrical casing at a clearance of about one eighth of an inch in the annular chamber formed therebetween. A supply of light lubricant normally occupies the lower portion of the annular chamber but rises to fill the chamber during rotation of the drum. The casing is enclosed in a housing, having a fan chamber containing an electric motor and fan or blower. The motor shaft may rotate both the fan and the drum.

8 Claims, 3 Drawing Figures

ORIGINAL PATENT UNAVAILABLE

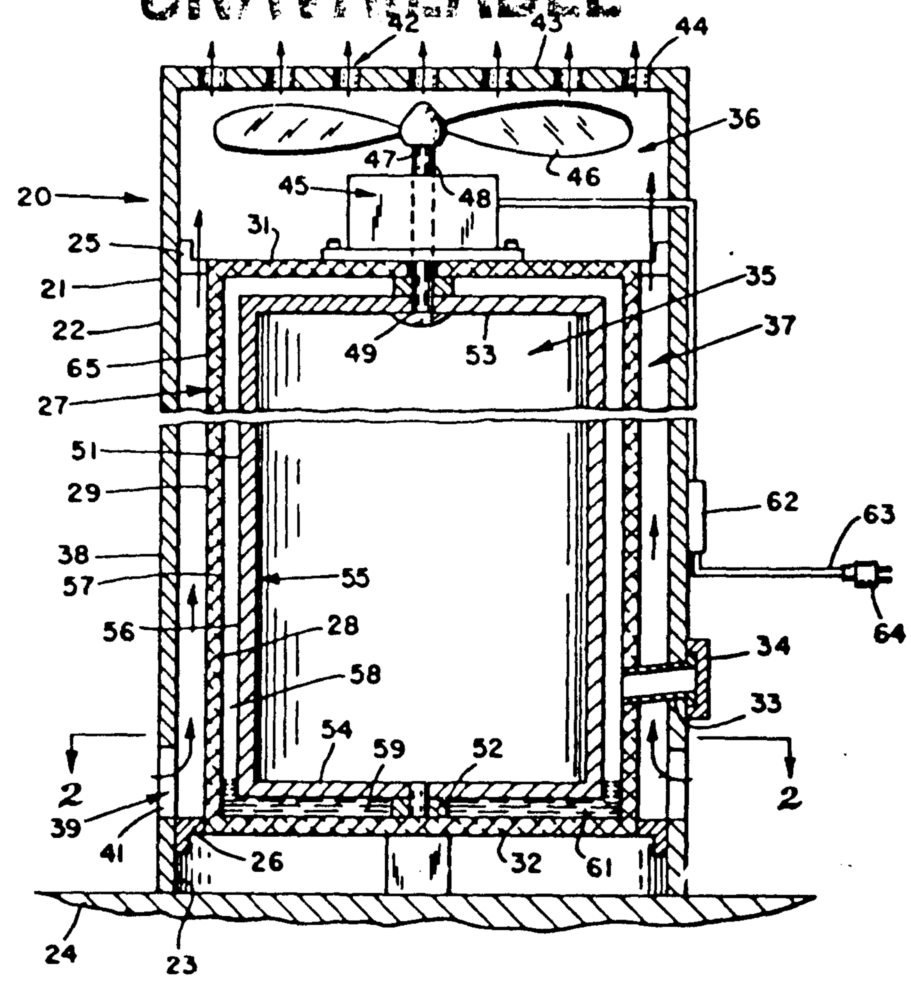
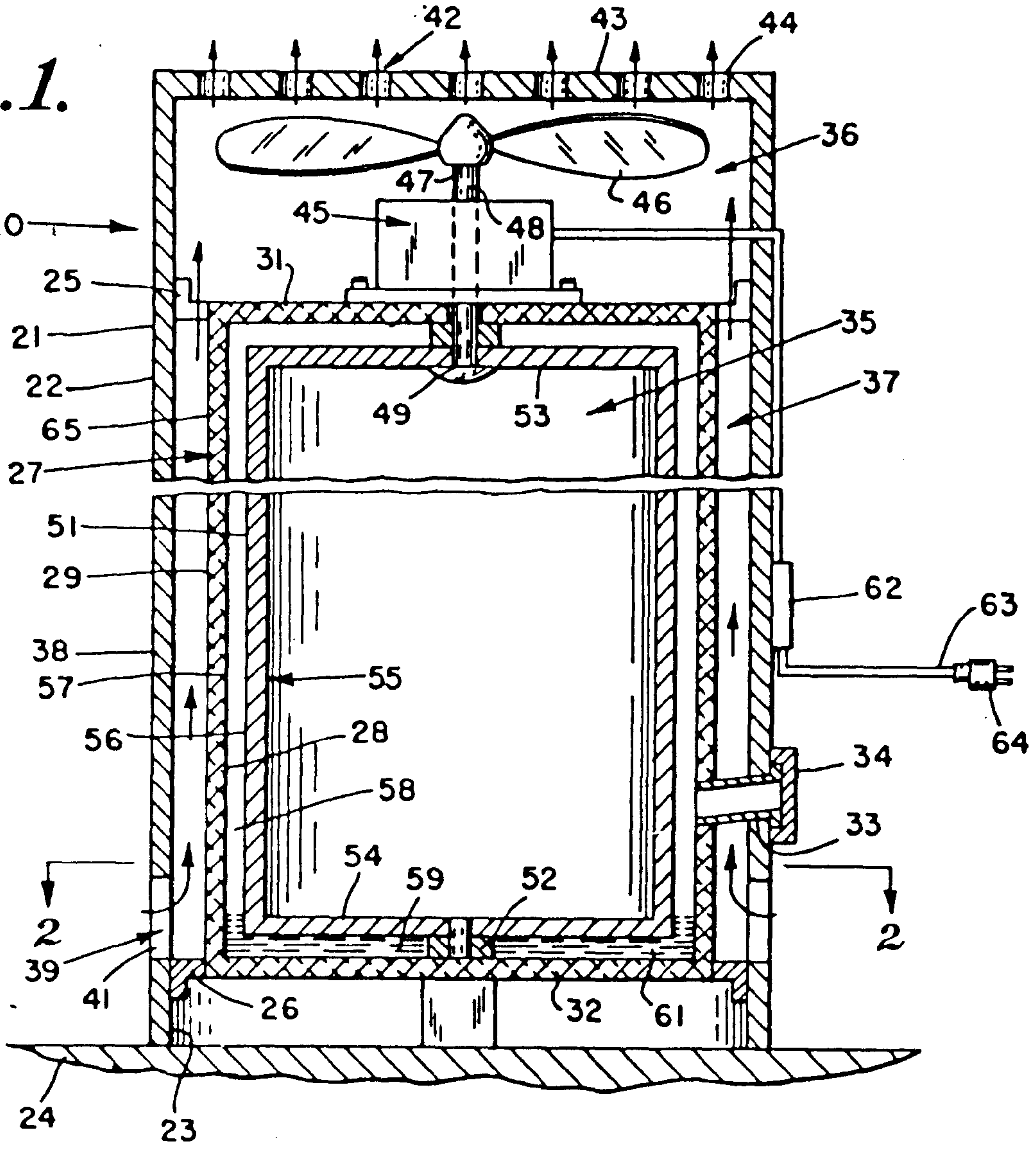


Fig.1.



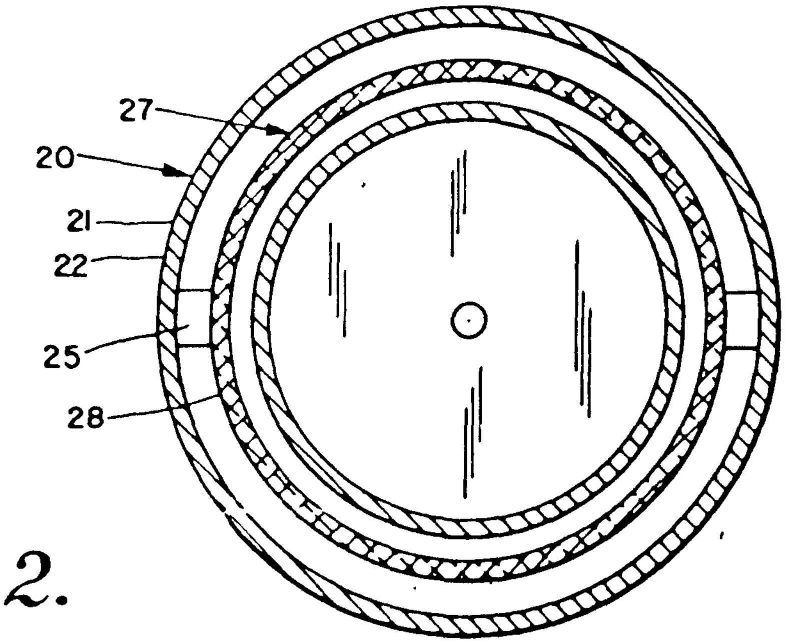
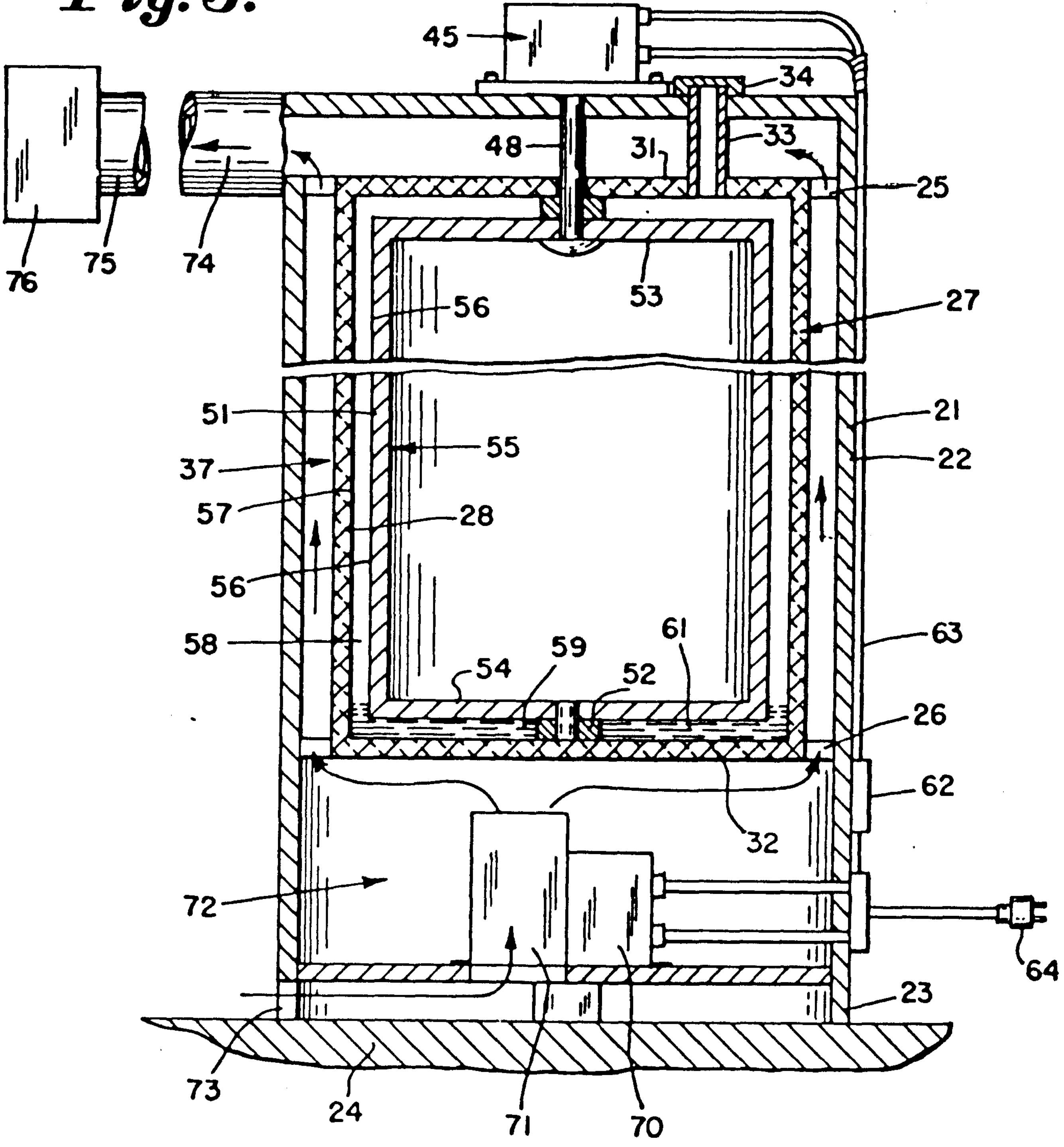


Fig.2.

Fig. 3.



FRICTION HEAT SPACE HEATER

BACKGROUND OF THE INVENTION

It has heretofore been proposed in U.S. Pat. No. 1,650,612 to Deniston of Nov. 29, 1927 to rotate a stack of discs relative to a coaxial stack of fixed discs on a horizontal axis within a casing to generate frictional heat in hot water flowing through the lower portion of the casing. In this heating device a supply of oil is contained in the upper portion of the casing to lubricate the discs and to float on the water at a predetermined level.

In U.S. Pat. No. 3,333,771 to Graham of Aug. 1, 1967, a pair of vaned rotors are each enclosed within a chamber of a casing, and mounted to rotate in a vertical plane on a horizontal axis as depicted in FIG. 7 thereof. As in the Deniston patent water flows through the device and is heated by friction.

In U.S. Pat. No. 4,004,553 to Stenstrom of Jan. 25, 1977 a single disc like rotor is revolved on a horizontal axis in a vertical plane, within a casing to heat water passing through the device.

SUMMARY OF THE INVENTION

Unlike the above mentioned patents wherein thin discs or vanes, in single or stack configuration, comprise the rotor, in this invention an elongated, cylindrical smooth surfaced, inner drum is the rotor. The drum is rotated in a horizontal plane on a vertical axis within an elongated cylindrical, smooth surfaced casing, or outer drum, to form an annular sealed, liquid, chamber therebetween having a clearance of about one eighth of an inch. A quart of relatively light oil is captive in the annular chamber and at rest occupies only the bottom thereof. However upon rotation of the drum, by an electric motor of about one horse power, the oil rises to fill the chamber due to the pumping action of the drum.

Thus friction heat is generated not by two metal, or other, surfaces contacting each other, but by the contact of the opposing surfaces with the oil which not only lubricates but generates heat.

A portable space heater is formed by enclosing the casing and drum in the lower chamber of a housing and drawing ambient air inwardly and around the heated outer surface of the casing for fan discharge back into the ambient atmosphere by a large diameter, eight bladed fan driven by the drum motor, or preferably by a separate motor. For use as a furnace an air blower and separate electric motor blow ambient air around the casing for discharge into a heating system.

BRIEF DESCRIPTION OF THE DRAWING

FIG. 1 is a front elevational view of the portable space heater of the invention, in half section;

FIG. 2 is a top plan view in section on line 2—2 of FIG. 1; and

FIG. 3 is a view similar to FIG. 1 of the device of the invention in its preferred form.

DESCRIPTION OF A PREFERRED EMBODIMENT

FIGS. 1 and 2 illustrate one embodiment of the friction heat heater 20 of the invention which includes an upstanding, hollow, cylindrical housing 21 formed of imperforate sheet metal 22 and having legs 23 for supporting it on a floor 24 of a building. The space heater 20 is portable and in the portable embodiment illustrated in FIGS. 1 and 2 the housing 21 is of predetermined diameter of about twelve inches and of predetermined height of about thirty-two inches.

Fixed within housing 21 by suitable brackets 25 and 26 is a hollow cylindrical casing, or outer drum, 27 which is of predetermined diameter less than the diameter of the housing, such as ten inches, and is formed of aluminum sheeting 28 for efficient transfer of heat. The cylindrical side wall 29, top wall 31 and bottom wall 32 of casing 27 are imperforate to form a sealed enclosure except for the filler tube 33, which is closed by a removable threaded cap 34.

The casing 27 divides housing 21 into the lower air heating chamber 35, which it occupies and an upper fan chamber 36, there being an annular air chamber 37 formed between the cylindrical side wall 29 of the casing and the coaxial, concentric cylindrical side wall 38 of the housing 21.

Air inlet means 39 is provided in the lower portion of the housing 21 in the form of spaced apertures 41 extending around the cylindrical side wall 38 and air outlet means 42 is provided in the top 43 of the housing in the form of apertures 44. The annular air chamber 37 connects the air inlet means to the air outlet means of the fan chamber 36.

A reversible electric motor 45 is mounted in the fan chamber 36 with an eight bladed fan 46 fast on one end 47 of the motor shaft 48, each blade being of about 25° pitch and the motor being about one horse power for rotating the shaft 48 at between 1800–3600 R.P.M.

The other end 49 of motor shaft 48 extends into the air heating chamber 35 to rotate the hollow, cylindrical drum 51 which is supported in suitable bearings 52 for rotating around the central, vertical axis of the casing 27 and housing 21.

The inner drum 51 is sealed and hollow and includes the top wall 53, bottom wall 54 and cylindrical side wall 55, the walls being of stainless steel. The exterior cylindrical surface 56 of the cylindrical side wall 55 is smooth as is the interior, cylindrical surface 57 of the aluminum of the cylindrical side wall 29 of casing 27 and the surfaces 56 and 57 are at about one eight inch clearance from each other to form a narrow, annular liquid receptacle 58 therebetween.

It should be noted that the annular liquid receptacle 58 is not a passage through which liquid to be heated is continually flowed, as in the above mentioned prior art patents. Instead it is a sealed chamber and is provided with a supply of liquid lubricant 59 such as a quart of No. 10 oil which normally rests in the horizontal space, or shallow liquid receptacle 61 between the bottom wall 54 of the drum 51 and the bottom wall 32 of the casing 27.

It has been found that the best results are obtained when the lubricant 59 is Quaker State F-L-M-A-T Fluid, Ford Motor Company Qualifications No. 2P-670306 M 2633F. Unlike prior patents, no water is in contact with the oil.

The motor 45 is connected to a thermostat 62, of any well known type by cord 63 and to a source of electricity by male plug 64 so that it is energized under the control of ambient temperature by the signals of the thermostat.

In operation the motor 45 drives the drum 51 at a substantial speed, which causes the oil 59 to rise up into the annular liqud receptacle 58 to substantially fill the same. The heat of friction between the inner drum 51 and outer drum, or casing 27 is transferred by the oil while it prevents wear on the surfaces 56 and 57 so that the exterior aluminum surface 65 of the fixed outer drum 27 becomes heated. Meanwhile the large diameter, multibladed fan 46 is drawing ambient air through the air inlet means 39, thence up through the annular air chamber 37 and past the elongated heated surface 65 for discharge through the air outlet means 42 back into the room.

As shown in FIG. 3, it is preferable to provide a separate electric motor 70, usually about ¼ H.P. and driving an air blower 71, these being mounted in a lower air chamber 72 for driving ambient air upwardly in an annular flow path in chamber 37 from the air inlet means 73 to the air outlet means 74. Air outlet means is the intake duct 75 of a hot air heating system 76 so that the heater 20 becomes a furnace rather than a space heater, the separate electric motor 70 enables the thermostat 62 to initiate rotation of the drum until a predetermined temperature is reached in the aluminum outer drum 27, whereupon the thermostat automatically de-energizes the drum motor 45 while continuing to rotate the separate fan, or flower motor such as 70, to furnish hot air to the room or heating system 76 until the casing 27 cools to a predetermined temperature.

I claim:

1. An odor-less, combustion-less, quiet beater comprising:

an inner, elongated, upstanding, hollow drum journalled within an outer, elongated, upstanding, hollow drum journalled within an outer, elongated upstanding drum to rotate in a horizontal plane on a vertical axis;

said inner drum having an outer, circumferential face spaced from the inner circumferential face of said outer drum by a fixed clearance of about one eighth of an inch to define an annular liquid receptacle therewith;

said inner drum having a lower diametrical face spaced from the lower diametrical face of said outer drum to define a shallow liquid receptacle therewith;

a supply of oil normally occupying only said shallow liquid receptacle but adapted to be raised upwardly into said annular liquid receptacle when said inner drum is rotated on said vertical axis at substantial speed;

and electric motor means for rotating said inner drum on said vertical axis within said outer drum to heat said oil and transfer heat to the outer face of said outer drum.

2. A heater as specified in claim 1 wherein:

said inner drum and said outer drum are both cylindrical in configuration.

3. A heater as specified in claim 1 wherein said inner drum is a sealed hollow cylinder and said outer drum is a sealed hollow cylinder except for an oil conduit connecting said annular liquid receptacle and said shallow liquid receptacle to outside said outer drum.

4. A heater as specified in claim 1 plus:

housing means extending around said outer drum, including an air inlet and an air outlet and

an electric motor operated fan within said housing for drawing air into said inlet, circulating said air around said outer drum and blowing air out of said outlet.

5. An odor-less, flameless, noiseless, heater comprising:

an inner, cylindrical, sealed, hollow metal drum telescoped within an outer cylindrical, hollow, metal drum,

said drums having upstanding cylindrical side walls spaced apart with a fixed clearance of about one eighth of an inch to form an annular, vertical, liquid, receptacle and having bottom walls spaced apart a predetermined distance to form a shallow, horizontal liquid receptacle

an electric motor operably connected to one said drum to rotate the same on a vertical axis relative to the other at a substantial speed, and

a supply of oil normally occupying only said shallow liquid receptacle but rising into said annular liquid receptacle, when one said drum is so rotated, to transfer frictional heat to said drums.

6. A combination as specified in claim 5 wherein:

said oil is a light oil and is normally in contact with both the bottom wall of said inner drum and the bottom wall of said outer drum.

7. A combination as specified in claim 5 plus:

a housing extending around said telescoped drums and

electric motor operated fan means in said housing for drawing ambient air into said housing, around said drums and dischargeing it out of said housing.

8. The method of generating heat in an odorless, flame-less noiseless manner by means of an outer, stationary, member and an inner member, rotatably mounted within said outer member, there being a supply of oil within said outer member, said method comprising the steps of

forming said members as hollow drums, one telescoped within the other, and both upstanding vertically so that the inner drum rotates on a central vertical axis relative to the outer drum and so that there is a clearance space between drums of only about one eighth of an inch in width,

limiting the supply of oil within said outer drum to a predetermined quantity so that it normally occupies only the bottom of the said clearance space

and then rotating said inner drum at substantial speed within said outer drum to heat said oil, cause it to rise up into said clearance space and transfer heat to said outer drum.

* * * * *

Twenty-sixth Day of *June 1979*

DONALD W. BANNER

Commissioner of Patents and Traden

Stove Requires No Fuel

Since unveiling his invention to a local newspaper Frenette has had to schedule appointments for the many people who have stopped in to see it. Visitors have included politicians, engineers, scientists, representatives of power companies, representatives from the Small Business Administration, heating specialists and dozens of persons wanting to buy state franchises to manufacture and market the new-style stove.

Frenette says his stove has been called into question only by those who have not seen it. And those who have seen it have not agreed on how it works.

Physicists not only deem the claim untrue, they are incensed that the so-called friction furnace is getting so much attention, according to a recent story in New Hampshire Profiles magazine by Robert Sullivan. The story offers the following opinions on Frenette's fuelless furnace:

"Professor William Mosberg of the University of New Hampshire asserts that any machine trying to heat a medium-sized home on 2 hp of energy 'is against all basic laws of thermodynamics'. Mosberg has not seen the machine, but says 'of course I am skeptical'.

"Professor John Kidder, of Dartmouth, hasn't seen the furnace either, but he is more than skeptical. 'This is a complete hoax,' he says, then tempering that statement somewhat, 'Either it's a hoax or the person's deluded about what he's getting'.

"Kidder continues: 'You can't get energy out without putting energy in. He can't make energy, he can only transform it.'

incensed as a taxpayer if funds were approved for development of the furnace. He concludes that the machine as it has been described to him would produce heat equivalent to that of 2 hair dryers."

Meanwhile, Frenette — a former home builder who operated a highly successful asphalt business for 18 years until 2 heart attacks forced him into early retirement — and his franchise holders, along with a host of other supporters, remain undaunted. They envision producing upwards of 100 different models to take care of a wide variety of heating needs — such as to small apartments, small homes, big houses, stores, shops, factories, and other buildings. There could even be models which would make it feasible to heat swimming pools, extending the season to 12 months of the year with enclosures, says Frenette.

You'll be hearing more about the controversial stove if and when production models become available. Meanwhile, for more details, contact: FARM SHOW Followup, Frenette Furnace, Box 255, Derry, N.H 03038 (ph 603 432-8313).

[illegible] AMERICAN BUSINESS/MARCH, 1979

NewsReal Series

Super Steam Making Machine Eliminates Need for Boilers

A technology is available that can effectively eliminate any need for using natural gas to heat water, homes or industry. It is a machine that can solve many of the energy problems which became go glaringly apparent last winter.

The device is a steam generating machine invented by the late Karl Schaeffer of Chicago. Schaeffer's method of making super-heated steam is instantaneous and will economically eliminate all need for huge boilers and direct use of fossil fuels.

Schaeffer's electrically powered steam machine also makes all the millions of constantly burning pilot lights obsolete

Among the more than 500 applications for steam, the Schaeffer device can effectively heat a modern home with 20 per cent less electricity than is now required.

This machine makes super-heated steam from running cold water instantly and without burning up a great deal of energy.

Sounds impossible doesn't it?

Credibility has been one of the key problems faced by Sonaqua Inc., the small firm that is attempting to develop and license the late inventor's lifetime dream.

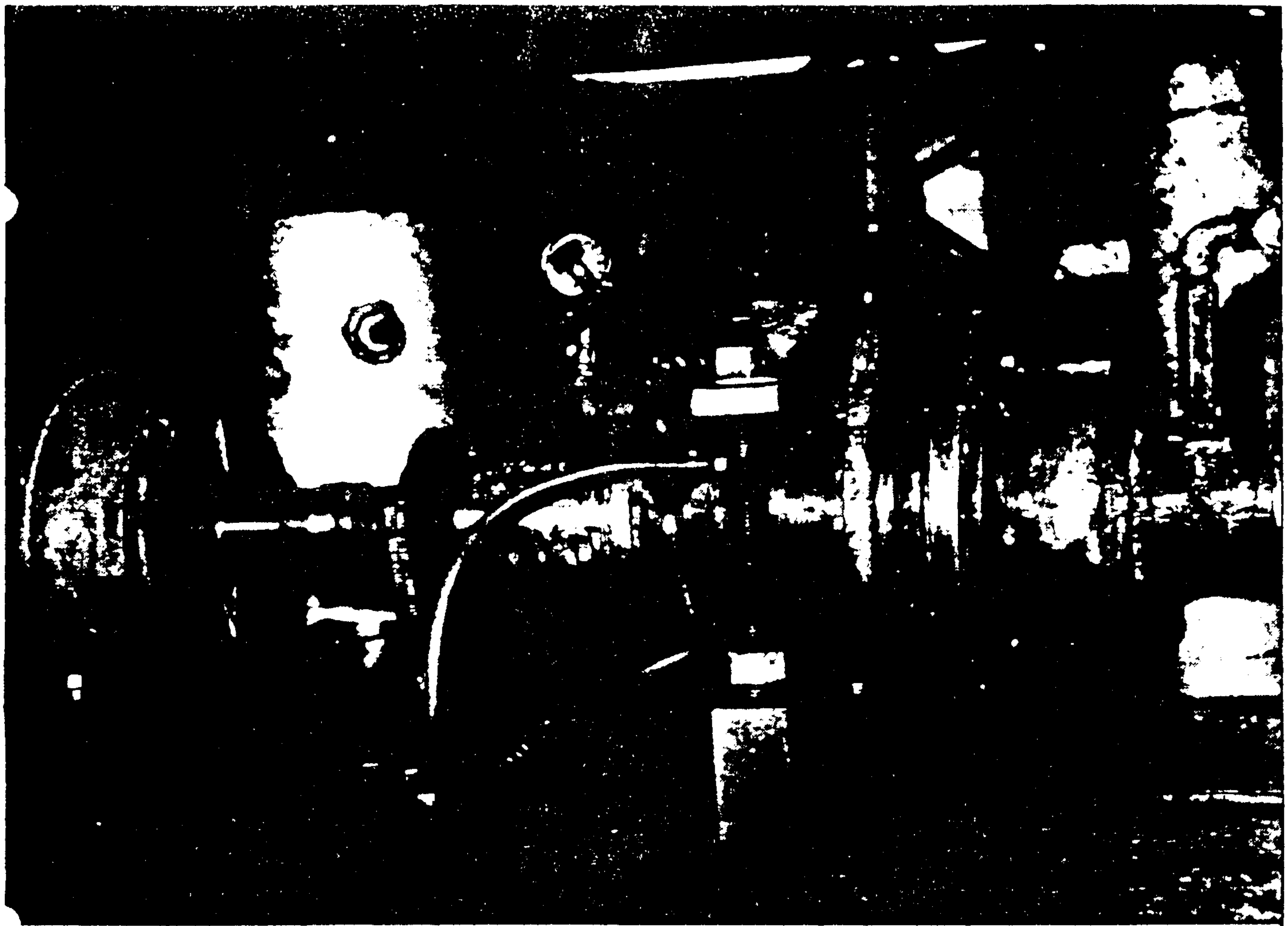
Engineers instinctively draw a blank when confronted with the claims of the Sonaqua group. Schaeffer accomplished something different; something not found in the text books.

Karl Schaeffer was 67 years old when this reporter first interviewed him an photographed his demonstration in 1973. The story was not told sooner because Sonaqua Inc. wanted the product fully developed before publicizing what they had.

The device is still not fully developed, but the story needs telling and our nation needs this new form of energy.

Karl Schaeffer found a way to harness some tremendous natural force which is inherent in molecules of water. There's no other way to state his principle.

"It all started for me when I was a



The unlikely looking aggregate of steel above is actually one of the greatest inventions to come along in the era of energy utilization. The device you see is a prototype of Karl Schaeffer's steam machine; it features a standard 20 horsepower electric motor (left) which drives a rotary assembly which shocks water into steam instantly – no boiler, no flame, no fossil fuels, no pilot light. Low cost steam heat is only one area of benefit.

graduating student of a trade-technical school in Berlin. The year was 1924," Schaeffer said in his thick German accent.

"I was in the washroom and when I turned off the water I heard the pipes knock," he said, referring to what engineers call "water hammer," a phenomenon in pipes that technology tries to eliminate.

"It was a loud knock and my mind suddenly said -- there is energy in that water hammer! And from that moment on, I was hooked."

"Of course I wondered if there was a way to harness such an energy, and I began my life-long quest -- well, it took me 50 years, but I have harnessed that energy and it can power the world.

"My machine can actually run forever so long as water continues to flow into it. Unlimited steampower! Free energy! Think of it!"

Karl Schaeffer was excused by his aides for "exaggeration." "Exaggeration my foot," Schaeffer bellowed. "I know what I have seen!"

The inventor had an unexplained engineering phenomenon occur four seperate times during the years he was striving to perfect his device. He turned off the electric power to the motor, but the machine kept running until the water in the tank was gone.

Enough power sprang from within his mechanism to continue pouring forth steam from cold water and turn all the mechanical apparatus as well -- with the power off!

Those unexplained incidents led Schaeffer to make claims considered "wild" by others. But even if he was deluded, which does not appear to be the case, his machine is by far the most efficient means of making steam ever devised by man.

Karl Schaeffer did it the hard way -- but after coming to America from his native Germany and devoting all his time and a considerable family fortune to his "mechanical obsession," he finally harnessed that simple shock wave in water phenomenon.

"It appears that this device actually makes use of two principles engineers try to eliminate -- water hammer and cavitation," noted Dr. Tom Hunter, former professor of Mechanical Engineering at Illinois Institute of Technology when he watched a demonstration of the Schaeffer machine.

"The force in the water is always there -- I have learned to use it, to intensify and use it," Schaeffer said emphatically.

One expert in engineering, chemistry and physics while watching the Schaeffer machine spin and spew forth steam said:

"I believe he is releasing energy that is inherent in molecules of water and this puts us on the verge of a totally new concept of energy utilization."

Essentially it is apparent that the vibrational and shock-wave nature of water can be used as a source of additional energy.

The Schaeffer machine is run by an electric motor that spins a metal disc. Cold water runs into the spinning flywheel where specially designed chambers cause an extremely rapid series of shocks to occur -- literally shocking the water into superheated steam.

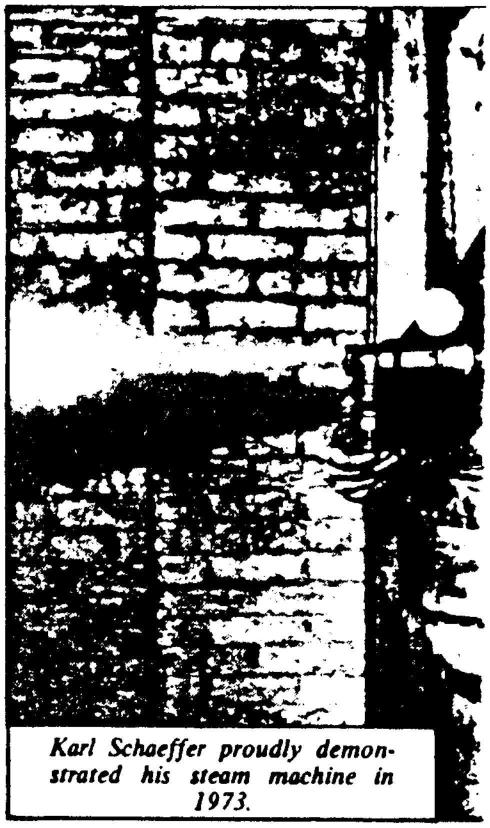
Schaeffer demonstrated this energy conversion device in his machine shop on Belmont Avenue in Chicago for this reporter a number of times. The device was crude and inefficient next to some later models made by Bob Price of Sonaqua, but it was impressive nonetheless.

"You just watch what happens," the old man said proudly as he stepped up to his floor mounted machinery. "You know how long it takes to boil water on your stove don't you?" he asked.

I nodded that I understood a little about boilers, heaters and making steam.

"Feel this," he ordered and I put my hand on some pipes leading into the device from a water tank. It was cold -- tap water cold.

A 20 horsepower electric motor was mounted beneath the heavy metal rotary impeller. A pipe protruded from the opposite side of the disc and bent up and out a window.



Karl Schaeffer proudly demonstrated his steam machine in 1973.

Schaeffer pulled a switch and I witnessed an amazing energy conversion -- something totally new to the annals of engineering. The motor whirred and Schaeffer tinkered with a valve for about 20 seconds, then before half a minute had elapsed superheated steam spewed forth and cascaded out the window.

The outlet pipe was too hot to touch in a very short time, yet the inlet pipe continued to be as cold as the water running inside it.

Indeed, a 20 horsepower electric motor is an energy source, but later tests and more sophisticated equipment showed that the power out -- in the form of steam -- was greater than the power in from the electric outlet source -- an engineering impossibility! ut a fact nonetheless.

In September of 1973 Sonaqua Inc. took the Schaeffer device to famed Battelle research institute in Columbus, Ohio for efficiency tests.

The test results, submitted to George W. Moffit Jr., board chairman of Sonaqua Inc. on Oct.4, 1973 by L.J. Flanigan of the Nuclear and Flow Systems Section of Battelle, listed the efficiency of the device over eight test runs to be in a range between 97.3 per cent and 99.0 per cent.

An interesting aside to the formal report is that much of the experimentation was said to have been "not definitive because shortcomings in the experimental apparatus introduced large uncertainties into the results."

However, it was reported by Battelle personnel, that part of that same unexplained phenomenon reported by the inventor must have developed: "We had readings in excess of 100 per cent on several occasions," an engineer told Bob Price, "and that's not possible."

"I was told that one of the readings actually indicated an efficiency of 117 er cent," Price said, trying to subdue a in at the thought of the confusion such a development must have caused the engineers.

The Sonaqua people, especially Price, are careful to avoid any claims for the unexplained energy at this time. "We have the greatest efficiency and potentially the lowest cost apparatus for making steam there is," Price said.

Sonaqua did manage to sell a licensee the right to produce home heaters, only to find that after two years the licensee is having the same developmental problems faced by the parent company - the new departure is up against dogmatic slide-rule skepticism and development money is hard to find.

Aquasonics of Denver has proven that a three bedroom home with a basement can be comfortably heated with two small three horsepower motors.

"They have managed to heat a home with 20 per cent less electricity than that used by a standard emersion unit - and they have had no trouble keeping the house warm --excessively warm," Price said.

For two winters now a model house in the chilly mountain city has had more than adequate heat from the two small motors and the Schaeffer device.

Sonaqua is applying for grants to study exactly what it is that makes the shocked water heat up. Schaeffer's sons, Kurt and Karl, have worked with Bob Price and have designed some experiments to help discover precisely what takes place within their father's mechanical device.

Whether grants for such studies are forthcoming or not, it seems silly to have the use of the device withheld. There's no doubt that it works. Time enough to figure out *why* after it's in production and helping to curb America's excessive energy use.

Edison's light bulbs were giving light when electric theory was in its infancy. It's not necessary to know precisely what is going on in Schaeffer's machine - the machine works consistently and that's what is needed.

Dr. Hunter, who viewed the device in action in this reporter's presence, said that it obviously performed in a wide range of efficiencies and eventually when the device was properly developed and finely tuned, the prospects were excellent for the product.

The professor of mechanical engineering did not care to comment on the "unexplained" part of the story.

Look at it this way. If the machine merely produces steam with a 98 per cent efficiency (which Battelle grants unequivocally) then its the best thing for home use yet developed.

And there's no need for large hot water storage tanks as smaller storage units, heated more efficiently will suffice; no need for pilot lights and natural gas.

In large apartment units there's no need for expensive boiler systems and either gas or coal fuel.

And if, when the skepticism is worn down and serious thought is given the device, the "unexplained phenomenon of the release of energy inherent in the molecules of water" is eventually proved and controlled, the world will have an alternative energy source that is sorely needed.

Hydro Dynamics' Steam Pump

by Tom Gibson

all images courtesy of [illegible]

Eleven years ago, Jim Griggs was working on an energy conservation project at a manufacturing plant in Missouri when he noticed something strange. A cold water supply line to a heating system felt warm to the touch.

When he asked a maintenance worker about it, the man said the plant had experienced water hammer in its pipes—the same force that rattles the pipes in your house when you turn off a kitchen faucet—and the cold water pipes became warm when it occurred. Intrigued by this, Griggs checked his engineering textbooks and confirmed that when a shock wave, such as one caused by water hammer, passes through a liquid, energy dissipates as heat.

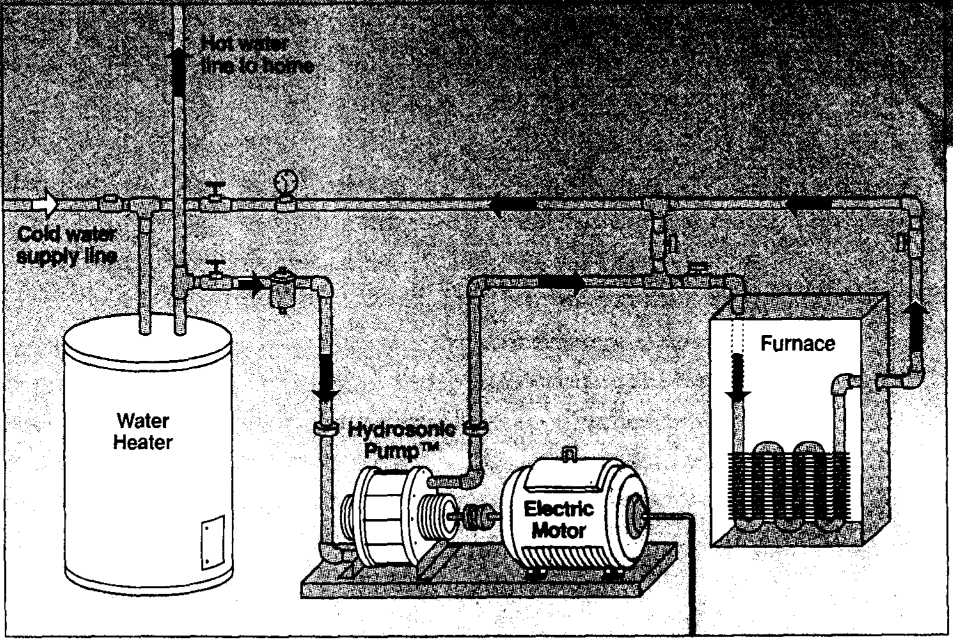
One engineer's problems—cavitation and water hammer—are another engineer's power source.

Most engineers would file away this tidbit of knowledge and think no more about it, but Griggs pursued it further—much further. He went on to develop a device called a steam pump that takes advantage of water hammer and cavitation—two phenomena engineers normally go to great lengths to avoid—to produce hot water and steam. He then started Hydro Dynamics, Inc., in Rome, GA, to develop and sell steam pumps. Today, after spending years perfecting his invention, Griggs stands poised to revolutionize the way we heat water for many applications and to create a spectrum of new uses for hot water and steam.

A Steam What?

An electrical engineer from Rockmart, GA, Griggs owned an engineering firm specializing in energy conservation and utility cost reductions when he started his adventure with steam pumps in 1986. He began by tinkering part-time in his garage, first trying a piston design, which proved unsuccessful. Beginning in 1988, he explored a rotary design. After trying hundreds of rotor configurations, he built the first successful prototype in 1989.

Structurally, the steam pump resembles an electric motor or centrifugal pump, in that a steel rotor rotates on a shaft inside a cylindrical steel housing. The specially designed rotor has hundreds of cavities drilled into it in a specific pattern. Cold water flows in one side of the housing, and hot water or steam flows out the other. Normally driven by an electric motor, the steam pump quickly produces hot water or steam, or a combination of the two, and it reportedly can generate steam at any pressure or temperature desired—all without combustion.



Typical hot water heating system.

After developing the working prototype, Griggs went full-time into steam pumps and started Hydro Dynamics in 1990. Kelly Hudson came on board in 1993 as president and CEO of the company, bringing with him business savvy developed from founding a company that makes carpeting out of plastic soda bottles. He had sold that business and retired at 46—but after only two weeks, he found he could not sit still. Hudson acknowledges that generating steam has little in common with weaving carpet, and he admits he actually found steam boring at the outset. But when Griggs explained the applications of the steam pump, Hudson became intrigued with the possibilities, and his thirst for starting a new business took over.

Research-and-Development Challenges

Hydro Dynamics has spent the last six years in research-and-development mode, building rotors by trial and error and testing them in field trial units. Griggs estimates he has built thousands of rotor designs over the last seven to eight years. By testing so many variations, he hopes to develop computer models of a steam pump, allowing future automation in the design process.

As one of its first trial units, Hydro Dynamics installed a 75-boiler-horsepower (BHP) unit at Rome Cleaners, a retail dry cleaner in Rome, GA. The steam pump heats water for use in cleaning operations and serves as a backup to the company's gas-powered boiler. Manager Ken Sledge reports, "I had my doubts at first as they were perfecting it, but it works. In five minutes you have steam pressure—it takes 30 minutes for our boiler."

The steam pump also takes up less space than a boiler, and can be used by customers, such as schools, nursing homes, and hospitals, that don't have a boiler room and lack boiler operators and maintenance personnel.

To boost the development effort, Griggs signed a Space Act Agreement in the early 1990s with NASA, which had an interest in the pump for space applications. Knowing it produced no sparks, NASA envisioned using the steam pump for distilling urine and other waste products as well as for converting cryogenic liquids to gases. In exchange for the opportunity to explore uses of the pump, NASA scientists provided technical assistance in analyzing the pump.

The space agency's expertise paid off in solving a problem with the rotor shaft bearings. As the pump heated up during water heating, the shaft would grow axially, stressing the bearings. NASA engineers came up with a pilot bearing design with increased clearances in the races.

When Jim Griggs originally developed his steam pump, he knew that water hammer caused it to heat water, but the actual mechanisms at work proved a mystery. Working with NASA also changed that.

As Griggs now confidently states, "The pump creates millions of short-distance shock waves which, in turn, cause a type of cavitation—controlled cavitation—inside the bores of the rotor. As the bubbles collapse, that's where the heat generation takes place." The rotation creates centripetal force and pressure differential, pushing water into the holes and causing shock waves.

The cavitation is the same phenomenon that occurs when a centrifugal pump tries to push more fluid out than it takes in, starving, in effect, the pump's inlet. Microscopic bubbles form in the water and implode as pressure increases in the pump. While these microbursts erode centrifugal pump impellers, Griggs has avoided such problems in the steam pump by virtue of the fact that the bubbles burst in the rotor pockets, away from the walls.

The energy output of a steam pump depends on many factors, including the size of the rotor, number of cavities, angle of the cavities, tolerances in the unit, and rotational speed. Griggs says the efficiency of the device has surprised many engineers. As examples of sizes, Griggs sees commercial applications extending to 150 BHP, and pulp and paper industry applications ranging from 1,000 to 10,000 BHP. Units can be stacked in modular fashion to increase horsepower.

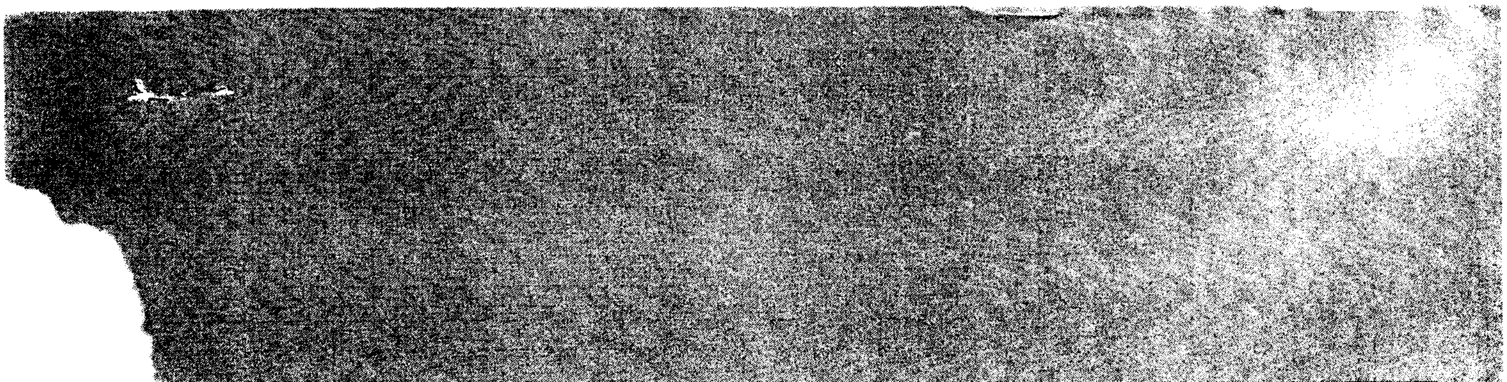
Griggs encountered another problem early on—finding a facility for testing larger versions of the steam pump. The closest place he could find with a test stand and electric motors large enough to turn the pumps was MJ Electric Enterprise, a motor repair and rebuilding service in Princeton, WV.

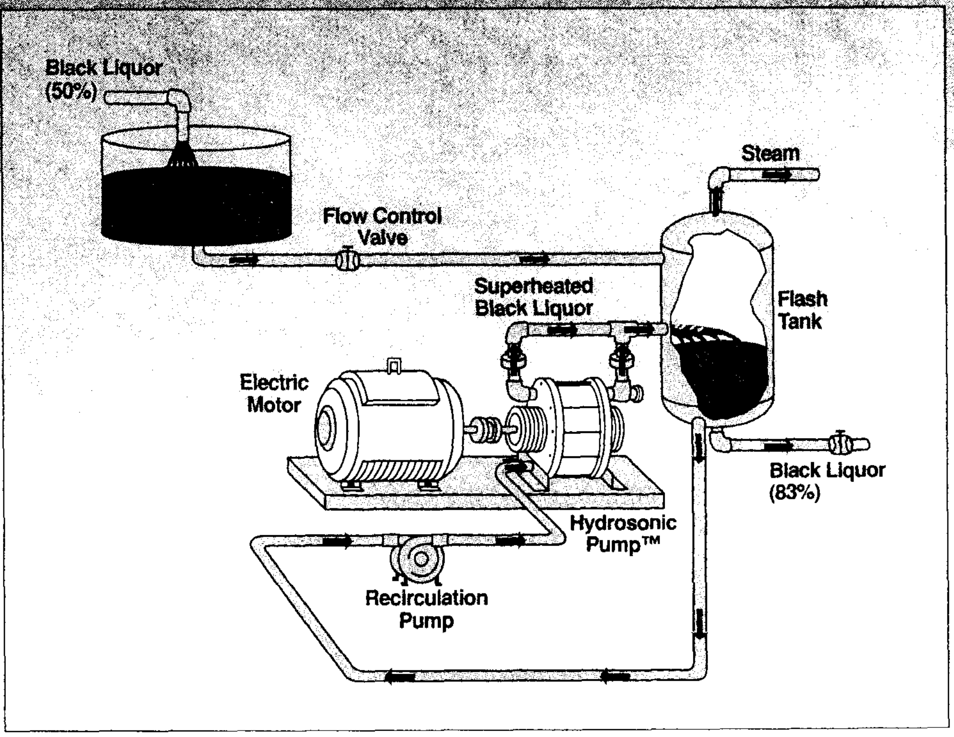
Perhaps the greatest problem Griggs has had to overcome is skepticism. As might be expected with a radically new technology, Griggs has encountered a mixed response in introducing steam pumps to the market. "It's been real hard in some industries to convince people that this technology will work. In others, the engineers have been very open to it."

One of Hydro Dynamic's biggest believers is Isolyser Company of Norcross, GA. The company manufactures products used in hospital operating rooms, such as surgical gowns, shoe covers, and masks, at its plant in Charlotte, NC. They make these products from OREX, a polymer made from polyvinyl alcohol. This unique material dissolves when agitated in hot water, allowing for easy disposal down a drain. Isolyser sells washing-machine-like processing units that agitate the OREX products at 205 °F for 20 minutes, and the company uses 40- to 50-BHP Hydro Dynamics steam pumps to supply the hot water in some situations. Alex Fairley, vice president and director of operations at Isolyser's Charlotte plant, says, "The thing we like about it [the steam pump] is it's small, and we don't have to worry about boiler codes."

Good for More Than Heating Water

As well as the steam pump works for heating water, Griggs thinks it offers greater promise for many other uses. "The really unique thing about the pump, compared to conventional boilers or heating mechanisms, is that you can run impure liquids through it without related scaling, coking, and similar





Hydrosonic Pump™ Concentrator System.

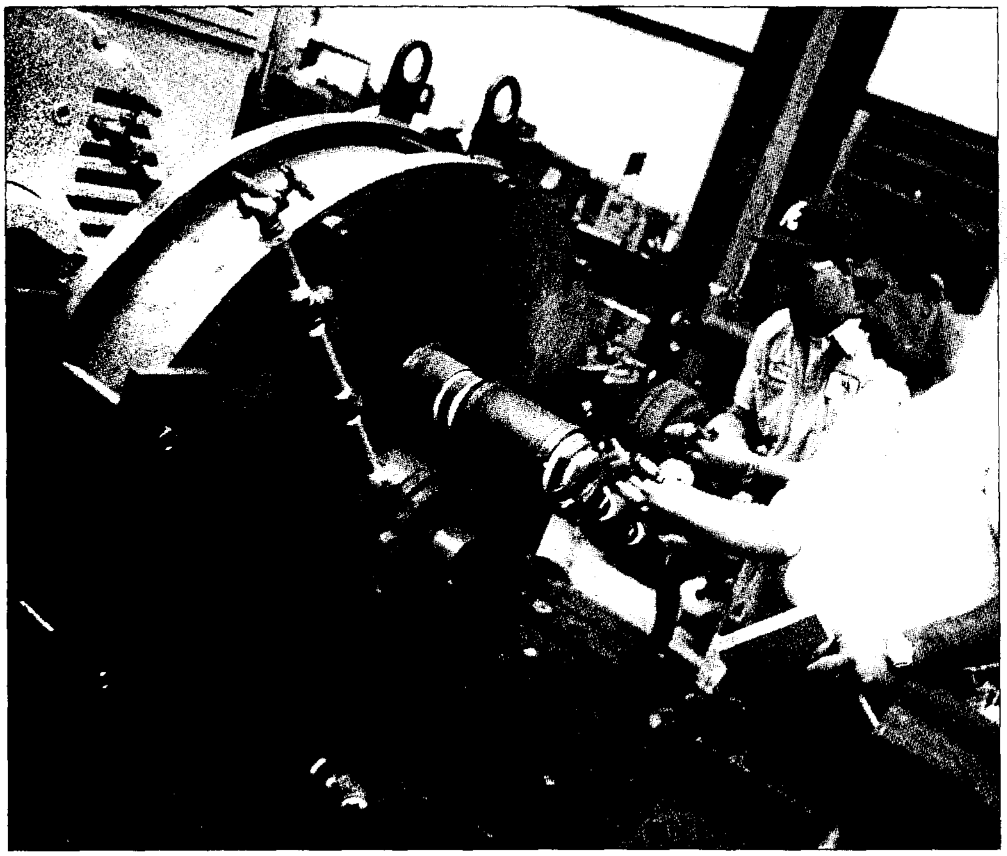
chemical problems." No buildup occurs because the temperature of the liquid and the steel surfaces inside the pump are the same when they come in contact (in contrast, consider what happens when a cold liquid hits a sizzling frying pan). And the rotor is inherently self-cleaning. "The environmental aspect of this could be huge," says Hudson.

This lack of scaling allows the pump to be used to distill—separate solids from liquids—many types of dirty liquids, resulting in reduced transportation and disposal costs, and allowing reuse of the liquid. In one of the first distillation applications, the pump removed dyes from discharge water at carpet plants in Dalton, GA. Another application involves separating dirt from the glycol de-icing fluid used on airplane wings so the fluid can be reclaimed. In the food processing industry, the pump can be used in homogenizing and pasteurizing milk. As Hudson says, "If it flows, we can run it through there."

"We think the most profitable field for us in the immediate future is going to be the pulp and paper industry," predicts Griggs. Paper plants use evaporators to separate wastes from liquids, but these devices experience scale buildup, which lowers their efficiency. Typically, a plant will have two evaporators, using one while cleaning the other. One papermaking process generates a gray sludge byproduct, a gooey mix of cellulose and fibers consisting of 2 percent solids and 98 percent water. Another process for cooking wood chips in a pulping operation yields a black liquor solution of 10 percent solids. In both cases, the steam pump can separate the solids. "We can actually get it dry enough so it can be burned in a furnace rather than landfilled," says Hudson.

Griggs and Hudson like to point out that the steam pump's potential extends to exotic areas, too. "Because the device is mechanically driven, we can have any drive source for this thing. We've calculated it running off everything from a turbocharger at a power plant to an ox walking around around a circle in some third world country," says Griggs. The steam pump reportedly can distill dirty river water or even saltwater for drinking. "One of my pet ideas is to direct-drive it with a windmill using ocean water as the fuel source and provide drinking water and electricity as a byproduct," continues Griggs.

Because the steam pump does not require a standard source of electricity, Hydro Dynamics



Steam pump inventor Jim Griggs (foreground) works with technicians at MJ Electric Enterprise prior to running a test on a 75-BHP steam pump to generate hot water.

plans to build portable versions for use in remote areas. These could see duty in distant environmental cleanup efforts. For example, a steam pump could generate steam from river or ocean water for cleaning up a beach after an oil spill.

Looking to the Future

Hydro Dynamics currently has 25 to 30 units in the field, but Griggs indicates the company will soon begin an aggressive marketing campaign that he foresees bringing in $8 million to $10 million in sales next year. The company has four salespeople in the field, including one in Europe.

The company operates out of a 20,000-square-foot facility, where eight full-time employees conduct research and development, as well as fabricate smaller steam pumps up to 20 BHP. The small plant has the capacity to manufacture 50 to 100 units a year.

Griggs does not envision Hydro Dynamics becoming a large manufacturer of steam pumps. He plans to license the technology to other companies. Hydro Dynamics will collect licensing fees or royalties and remain a small research-and-development firm. Typically, licensees will consist of manufacturers of equipment sold to end users, for example, makers of concentrators and evaporators for the pulp and paper industry. MJ Electric also has expressed an interest in producing units, as they see steam pumps as similar to electric motors in a fabrication sense.

Meanwhile, Hydro Dynamics continues to explore new applications for its steam pump. Realizing they have only begun to tap the device's potential, Griggs says, "We're always coming up with new applications, and Kelly and I both feel that the best use for this device probably hasn't been thought of yet."

For More Information

To learn more about the steam pump, write to Hydro Dynamics, Inc., 8 Redmond Court, Rome, GA 30165, U.S.A.; or call: (706) 234-4111; fax: (706) 234-0702. CA

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The James Griggs Hydrosonic Pump

By Joe Maize

Editor's Note: James Griggs is a speaker at this year's International Symposium on New Energy to be held in Denver April 25-28, 1996.

James Griggs is an electrical engineer, entrepreneur, and owner of Hydro Dynamics, Inc. located in Rome, Georgia. Mr. Griggs has invented what is called a hydrosonic pump, which is a novel device for heating liquids. This device, rather than heating liquids via a resistive heating element or via the direct combustion of fossil fuels, heats liquids through what may be called shock waves, hence the name hydro+sonic (water + sound). In watching the video of this hydrosonic pump, one sees cold water being injected into one side of this device while saturated steam exits the other side of this device. Beneath the external casing of the device itself, unseen, is a spinning cylinder of novel design. Impressive as this device is, it is more so when it is learned that the energy of the steam exiting this device greatly exceeds all of the energy input needed to create this steam. In other words, the energy efficiency of this device is significantly greater than 100%. Equally impressive is the fact that the steam exiting the hydrosonic pump glowed violet.

The history of this product's development is as interesting as its performance and perhaps has antecedent developments in the discovery of sonoluminiscence in the 1930s and the development of the Keeley motor by John Keeley in the late 1800s. Sonoluminescence is the direct conversion of sound into light using water as the medium, hence the name sonoluminescence (sound + light). Sonoluminescence is not well understood even to this day, 60 years after its discovery, but apparently it is the effect of light emission from a bubble dissolved water as a result of a shock wave created in that water by sound waves which hammer the bubble into compression. The bubble releases energy over a broad spectrum of visible as well as invisible light, the most energetic components being in the blue and violet spectrum. The Keeley motor was an over unity device which ran on water and whose operation purportedly was based on "water hammer," that undesirable phenomenon in any high pressure water pipe system which describes the unwanted banging and groaning noises associated with a system due to abrupt pressure changes or shock waves.

In the mid to late 1980s, Jim Griggs toyed with a desire for alternative sources for energy production. Griggs was familiar with the water hammer effect and noticed that articles on this subject mentioned the accompanying heating effect, albeit a minuscule one, as a result of the creation of the water hammer shock waves. By magnifying this shock wave effect via a novel rotating cylinder design, Griggs hoped to magnify this heating effect as well. In 1988 and 1989, Griggs made his firs successful rotors which could convert water directly into steam. I was in the 1989 design that Griggs noted that his energy efficiency exceeded 100%. Griggs stated, "I felt we had made a mistake, but we could not make the results go away. Now here we are, five years later, and after many of our own tests, as well as calling outside consultants to run and evaluate the system objectively, the (over unity) results still stand." Even though the academic community has pretty much ignored the over unity aspects of the hydrosonic pump, the market place has not ignored the unique heating attributes of this pump. For instance, the pump is already being marketed for home and commercial heating systems as an

Continued on next page

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Test Results of an N-Machine

The following is a summary of the test results performed by Leyland Consultants Ltd. of Auckland as prepared in report form by Bryan Leyland. This test was a measurement of power efficiency and was performed for the prototype Space Power Generator (the SPG, Bruce DePalma's N-Machine) for Space Power Corporation Limited. Leyland Consultants is a consulting firm of electrical and mechanical engineers. Leyland consultants say they spent about an hour in discussions with Bruce DePalma to understand the differences between his machine (the SPG) and the standard homopolar generator, which must by conventional wisdom have a power efficiency less than 100%. The Leyland test was performed on December 20, 1994 at 10:35 a.m. The test ran for a total time duration of 13 minutes after which time it was determined that the total power efficiency (defined as total power output divided by total power input) for the Space Power Generator was 57% +/- 7.5%.

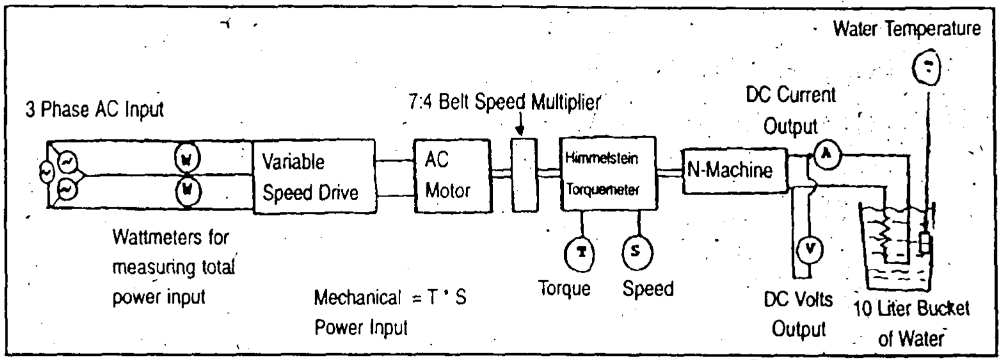


Figure 1: Test setup used to test the N-Machine (de Palma homopolar generator) on 12/20/94.

Although the actual test performed by Leyland Consultants only lasted 13 minutes, it certainly took much longer for them to set up the proper equipment before the test could be run. As shown in Figure 1, they used 2 wattmeters to measure the total power into their variable speed drive circuit, which then drove an AC induction motor, which then drove a belt speed multiplier, which then drove a Himmelstein Torquemeter, which then drove the SPG generator under test, which then dumped its power output as resistive electrical heat into a 10 liter bucket of water. The 57% efficiency number was obtained by dividing the power dumped into the bucket of water (2802 Watts) by the average input reading of the Himmelstein Torquemeter (4925 Watts). In criticizing their own results, Leyland Consultants noted that they did not have the capability of measuring the output d.c. current of the SPG N-Machine, which they estimated to be approximately 5448 amps based on a low amperage (1765 amps) measurement, which they could measure. Since there were obvious losses between the copper conductor output wires of the SPG and the bucket of water, which could easily have been 100 Watts or more, Leyland suggests that for future testing either a 6000 amp ammeter be used to measure output current accurately or more of the SPG output conductors be submerged in the water bucket to obtain more accurate calorimetric measurements. Using the higher range ammeter, one need only multiply the SPG output voltage, V, by its output current, I, to obtain the output power, assuming both V and I were constant.

A second criticism of their results, which Leyland noted, was the fact that there was torque ripple in their measurement for input power using the Himmelstein Torquemeter, hence detracting from the accuracy in measuring input power to the SPG (3 of 10 torque-speed data points were missing from their spreadsheet). Averaging the torque reading and multiplying this by the rotational speed would give an accurate, precise measurement of power delivered to the SPG only if there were no speed ripple. Measuring rotational speed and torque to calculate power is analogous to measuring voltage and current to calculate power. If the rotational speed were 90 degrees out of phase with the torque (just as when the voltage is 90 degrees out

Continued on next page

Griggs Pump continued

inexpensive steam generator. The dairy industry is currently interested in its use to pasteurize milk, which requires that milk be heated to a temperature of 160 degrees Fahrenheit for 16 seconds. A condominium complex and boiler company in Key Biscane, Florida is working with Griggs to generate steam directly from water using a windmill generator. In this case, the hydrosonic pump would be powered directly by the windmill. Other contacts in the Bahamas, Puerto Rico, and Canada are also interested in this development as well. Lastly, the hydrosonic pump is being used now industrially, in distillation processes. For instance, it is being used in one process to separate glycerin from water.

Unfortunately, Griggs must play down the over unity aspect of his invention for, as he says, ". . . fear of chasing a customer away with talk of a device that defies energy conservation laws." Of course, if this same customer were to see the violet luminescent steam exiting from his newly installed hydrosonic pump, one must wonder whether or not this customer would realize that he was witnessing something for which academic science does not allow. ▼

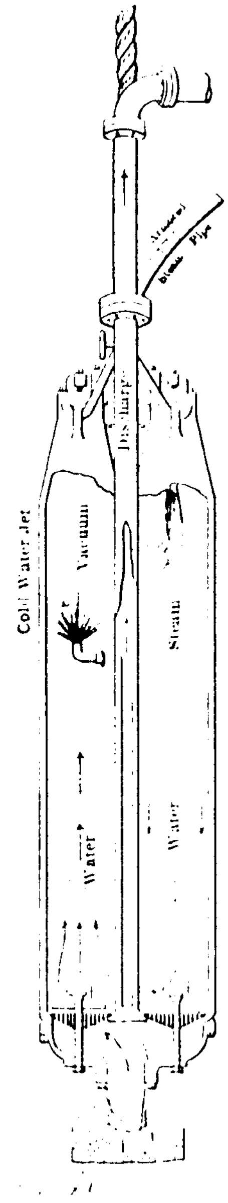
Engineering and Mining Journal.
Vol. 85. March 14, 1908. p. 555.

The Emerson Steam Pump

The pulsometer pump has many advantages of construction over other types of pumps when water containing grit is to be handled; it is therefore especially adapted to shaft work. In such pumps, the water comes in contact with only four valves, two in the suction and two in the discharge pipes. As these are ordinary, flat-seated, rubber valves, they do not wear and get out of order easily when gritty water is being pumped. Besides, in pulsometer pumps there are no stuffing boxes, no plungers, no pistons, nor other moving parts such as cause trouble in reciprocating pumps, used for pumping dirty water. A pulsometer pump is self-contained; the valve, controlling the admission of steam to the cylinders, is situated in the top of the pump where it is not reached by the gritty water and where it is protected by the body of the pump from pieces of flying rock during blasting.

Another advantage of the pulsometer pump is that it can be easily moved, owing to its not requiring any foundation setting. On account of the absence of reciprocating parts, there is little vibration of the pump and so it can be hung from a rope while working; this permits of easy hoisting or lowering. Besides, as the steam acts directly upon the water without the use of plunger or piston, the pump is generally somewhat lighter than a reciprocating pump of the same capacity. As pulsometer pumps do not exhaust the steam into the air the shaft is therefore much cooler than when a sinking pump of the ordinary type is used. Pulsometer pumps can be worked when nine-tenths submerged.

The great drawback of pulsometer pumps has been the large amount of steam consumed. In a new pump of this type, made by the Emerson Steam Pump Company, of Alexandria, Va., it is claimed that the steam consumption is reduced by means of an early cut-off, resulting from the special design of the rotary slide-valve steam ports. The Emerson pump, like other forms of pulsometer pumps, depends upon the vacuum in one chamber, produced by the condensation of the steam by means of a jet of water, to suck up the water into the chamber. Then by means of the rotary valve, steam is turned in on top of the water and the water is forced out of the pump by the steam itself.



EMERSON STEAM PUMP

On the Rand during the year ending June 30, 1907, there were 69 producing mines, using 8047 stamps which ran on the average 337.456 days. The tonnage hoisted amounted to 18,189,301 tons belt will run off a crowned face pulley more easily than from a straight-faced one. Different authorities recommend very different amounts of crowning. One authority recommends a crown or height at the center of 1/20 of the width of the pulley for leather belting, and 1/150 of the width for cotton belting. Another recommends from 1/16 to ⅛ inch of crown for each foot of width of pulley for high speeds, and ¼ inch of crown for low speeds. A formula which has proved satisfactory is as follows: 1/200 of pulley face $+$ 0.020 inch $=$ height of crown.

Pulley Lathe. The pulley lathe is a special design intended for turning and facing pulleys, gears, flywheels, couplings, and work of a similar nature. A well-known design of pulley lathe has two tool-rests, one located at the front and the other at the rear of the machine. The tool-rests are mounted on compound slides, one of which is utilized for diameter adjustment, and the other for feeding the tool when turning. These slides are carried by a table which swivels in the center and has a series of holes drilled on one side, through which a pin is inserted in order to locate the table at whatever angle may be required to give the desired taper or "crown" on the pulley. The tool-slides feed in opposite directions, so that each tool turns half of the pulley rim.

Pulley Speeds. The following rule for the relation between the sizes of pulleys and the number of revolutions of two shafts can be formulated: The number of revolutions of one shaft multiplied by the diameter of the pulley on the same shaft, divided by the number of revolutions of the second shaft, gives the diameter of the pulley on the second shaft. To obtain the speed of the driven pulley in a compound drive, proceed as follows: Divide product of diameters of driving pulleys by product of diameters of driven pulleys, and multiply quotient by speed of first driving pulley.

Pulley Taps. Originally designed for use in tapping set screw holes in the hubs of steel pulleys. The thread length is approximately equal to that of a regular (standard) hand tap, while the shank is full diameter of the thread and comes in various lengths. Recently the field for this tap has been expanded to meet the demands of those who want hand taps in longer lengths.

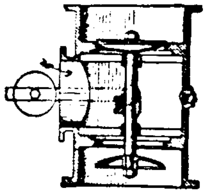
Pulsometer. The pulsometer, sometimes known as an aquometer, may be defined as a steam pump which acts partly by direct steam pressure and partly by vacuum. It has two working chambers into which the steam is alternately admitted. A partial vacuum is formed by the condensation of the steam, then water rushes into the chamber on account of the vacuum thus formed. When the chamber is full of water, the valve opens, steam enters again, and forces the water out into the delivery chamber, the steam condenses as before, causing the inflow of another supply of water, and the cycle is repeated. Two chambers are provided so that one is filled while the other is discharging. The pulsometer has neither rotating nor reciprocating parts. It will raise water to a height of about 26 feet, although it is not advisable to have a lift exceeding 20 feet, and it will force the water, if necessary, to a height of about 100 feet.

Pump. A pump may be defined as a mechanical device or machine designed for elevating or conveying liquids against the action of gravity, or for exhausting air or other gases from a closed vessel. A pump for liquids may be intended primarily for elevating the liquid from a source of supply below the pump up to the pump, or the principal purpose may be to force the liquid either to a much higher level or to some distant point by connecting the pump with suitable pipes. A mechanical device for withdrawing air from a closed vessel is ordinarily classified as a *pump*, but, if designed for compressing air or other gases, it is known as a *compressor* or *blower*.

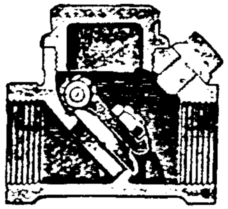
Pumps are classified either with reference to some characteristic constructional feature or the particular class of service for which they were designed. The common types of pumps may be divided into several general classes, as follows: 1. *Reciprocating pumps* or those having a piston or plunger which is given a reciprocating movement in the pump cylinder. 2. *Centrifugal pumps* having a rotary impeller which, as a result of centrifugal force, causes the water to flow from the center to the periphery of the impeller with increasing velocity, and then out through the discharge outlet. 3. *Rotary pumps* which, according to the general usage of the term, differ from the centrifugal pumps in that the water or other fluid is forced through the pump by the direct application of pressure from rotating pistons or impellers and independently of centrifugal action. 4. Pumps in which the fluid is moved by the direct application of steam or compressed air and without employing a reciprocating piston or rotating impeller. In each of these four general classes, there are many different types.

Pump, Air or Vacuum. See Air or Vacuum Pump.

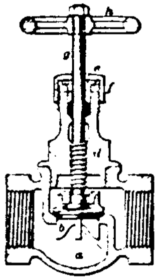
Pump Displacement. The displacement of a pump is equivalent to the "effective area" of the pump piston or plunger multiplied by the length of the stroke, or it is the volume included in a complete stroke. To determine the displacement in cubic inches per minute, multiply the effective area of the piston or plunger



417. AUTOMATIC RELIEF VALVE.—The valve is kept closed by a crank attachment to the spindle and weighted lever outside. Excess of pressure raises the stem and discs, throttling the passage of steam and relieving the back pressure.

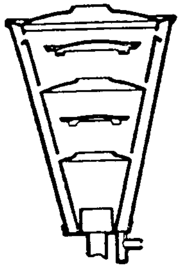


418. HORIZONTAL SWING CHECK VALVE.—The disc is loose in the swing frame and may be reground tight by a socket wrench passed through the plug opening.

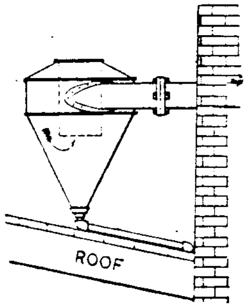


419. GLOBE VALVE.

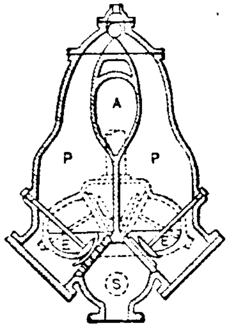
a, the body.
d, the bonnet.
g, the spindle.
b, the winged disc.
c, the spindle nut.
e, gland.
f, gland nut.
h, wheel.



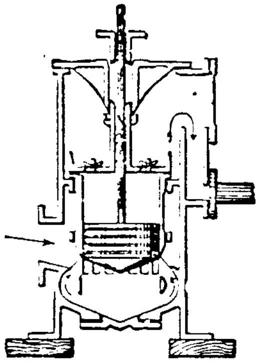
420. EXHAUST STEAM HEAD.—The exhaust steam is deflected by perforated discs and cap plates, which separate the water to drip between the inner and outer shell.



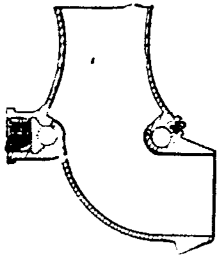
421. CENTRIFUGAL EXHAUST HEAD.—The exhaust steam head enters the drum tangentially, throwing the particles of water against the outer surface to drip to the bottom.



421*a*. THE PULSOMETER STEAM PUMP.—Water is forced from each chamber alternately by the steam pressure, while the opposite chamber is filled by the vacuum caused by the condensation of the steam in contact with the wet surface. The ball valve is very lightly balanced and is thrown over by the alternating vacuum and steam pressure.



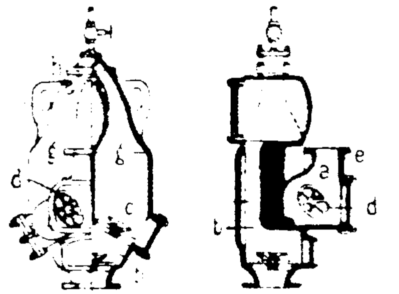
421*b*. THE EDWARDS AIR PUMP.—Has no suction valves. The ports in the cylinder are opened by passing the piston to the bottom of the cylinder. The water and air enter above the piston and is discharged above. The discharge valves are sealed by water held back by the dam. The piston rod is sealed by a water filled cup.



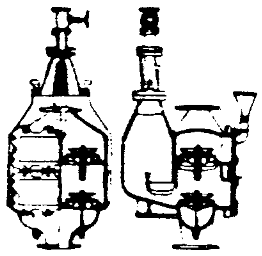
421*c*. STEAM SOOT SUCKER for cleaning boiler tubes by drawing the soot and ashes from the tubes by an annular steam jet.

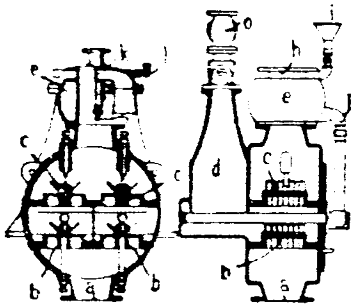
Mechanical Movements; Powers and Devices. Gardner D. Hiscox. 12th ed., 1911. p. 115

1	**g) Dampfdruckpumpen** (f pl), **Dampfpulser** (m pl), **Pulsometer** (n pl) **Steam Pressure Pumps** (pl) **Pompes** (f pl) **à pression de vapeur**		**Насосы** (m pl) **съ непосредственнымъ давленіемъ пара на жидкость** **Pompe** (f pl) **a pressione di vapore o prementi a vapore, pulsometri** (m pl) **Bombas** (f pl) **de vapor á presión, pulsómetros** (m pl)
2	kolbenlose Dampfpumpe (f), Dampfdruckpumpe (f), Dampfpulser (m), Pulsometer (n) pistonless or pulsating steam pump, pulsometer pompe (f) à vapeur sans piston, pulsomètre (m)		безпоршневой паровой насосъ (m); пульсометръ (m) pompa (f) a vapore senza stantuffo, pulsometro (m) bomba (f) de vapor sin émbolo, pulsómetro (m)



3	Neuhausscher Dampfpulser (m), Neuhaussches Pulsometer (n) Neuhaus' pulsometer pulsomètre (m) de Neuhaus		пульсометръ (m) системы Нейгауза pulsometro (m) di Neuhaus pulsómetro (m) de Neuhaus
4	Druckkasten (m) pressure chamber chambre (f) de refoulement	a	напорная камера (f) camera (f) premente e di pressione cámara (f) de presión
5	Saugraum (m) suction chamber chambre (f) d'aspiration	b	всасывающая камера (f); камера всасыванія camera (f) aspirante e di aspirazione cámara (f) de aspiración
6	Saugventil (n) suction valve clapet (m) ou soupape (f) d'aspiration	c	всасывающій клапанъ (m) valvola (f) aspirante válvula (f) de aspiración

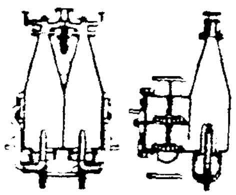
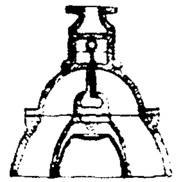
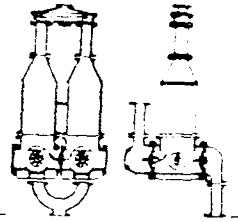
Druckventil (n) pressure valve soupape (f) de refoulement	d	нагнетательный клапанъ (m) valvola (f) premente válvula (f) de impulsión	1
Steigrohrstutzen (m) delivery pipe branch tubulure (f) de refoulement	e	патрубокъ (m) нагнетательной трубы attacco (m) del tubo montante extremo (m) de empalme del tubo de impulsión	2
Kammer (f) des Dampfpulsers, Pulsometerkammer (f) pulsometer chamber chambre (f) ou poche (f) du pulsomètre	g	камера (f) пульсометра camera (f) del pulsometro cámara (f) del pulsómetro	3
Luftventil (n) air valve valve (f) à air	h	воздушный клапанъ (m) valvola (f) dell'aria válvula (f) de aire	4
Körtingsches Normalpulsometer (n) Körting's normal pulsometer pulsomètre (m) normal, système Körting		нормальный пульсометръ (m) (системы Кертингъ) pulsometro (m) normale di Körting pulsómetro (m) normal de Körting	5



Körtingsches Hochdruckpulsometer (n) Körting's high pressure pulsometer pulsomètre (m) à haute pression, système Körting		пульсометръ (m) высокаго давленія (системы Кертингъ) pulsometro (m) ad alta pressione, sistema Körting pulsómetro (m) de alta presión de Körting	6
Saugstutzen (m) suction pipe branch tubulure (f) d'aspiration	a	всасывающій патрубокъ (m) attacco (m) del tubo aspirante extremo (m) de empalme de aspiración	7

Dictionnaire Illustre Termes Techniques en six langues; Vol. XII: Hydraulique, Pneumatic, Technique du Froid. ed. Alfred Schlomann. (Paris: Dunod, 1915). p. 268-271.

1	Saugventil (n) suction valve soupape (f) d'aspiration	b	всасывающій клапанъ (m) valvola (f) aspirante válvula (f) de aspiración
2	Druckventil (n) pressure *or* delivery valve soupape (f) de refoulement	c	нагнетательный клапанъ (m) valvola (f) premente válvula (f) de impulsión
3	Pulserkammer (f), Pulsometerkammer (f) pulsometer chamber chambre (f) *ou* poche (f) *ou* cloche (f) de pulsomètre	d	камера (f) пульсометра camera (f) del pulsometro cámara (f) del pulsometro
4	Einspritzwindkessel (m), [Injektionswindkessel (m)] air injection vessel *or* chamber cloche (f) d'air d'injection	e	инжекціонный воздушный колпакъ (m); воздушный колпакъ впрыскивателя cassa (f) d'aria d'iniezione cámara (f) de aire de inyección
5	Strahlrohr (n), [Injektionsrohr (n)] injection pipe tuyau (m) d'injection	g	инжекціонныя труба (f); труба впрыскивателя tubo (m) d'arrivo del vapore o d'iniezione tubo (m) de inyección
6	Druckstutzen (m) delivery pipe branch tubulure (f) de refoulement	h	нагнетательный патрубокъ (m) attacco (m) del tubo premente extremo (m) de empalme de impulsión
7	Fülltrichter (m) funnel for priming entonnoir (m) de remplissage	i	воронка (f) для наполненія imbuto (m) di riempimento embudo (m) de llenado
8	Steuerzunge (f) clapper valve languette (f) de distribution, obturateur (m)	k	управляющій или распредѣлительный язычёкъ (m) piastra (f) *o* linguetta (f) di distribuzione lengüeta (f) de distribución
9	Luftventil (n) air valve soupape (f) à air	l	воздушный клапанъ (m) valvola (f) d'entrata d'aria válvula (f) de aire
10	Dampfventil (n) steam valve soupape (f) d'admission de vapeur	o	паровой клапанъ (m) valvola (f) del vapore válvula (f) de vapor

doppelt wirkendes Pulsometer (n) double acting pulsometer pulsomètre (m) à double effet		пульсометръ (m) двойного дѣйствія pulsometro (m) a doppio effetto pulsómetro (m) de doble efecto	1
Pendelklappe (f) clapper [steam] valve languette (f) de distribution, obturateur (m)		качающаяся заслонка (f); маятниковый клапанъ (m), языкъ (m) съ верхнимъ подвѣсомъ valvola (f) od animella (f) o linguetta (f) a pendolo válvula (f) pendular, lengüeta (f) de distribución	2
Unterdruckdampfpumpe (f), [Dampfvakuumpumpe (f)] steam vacuum pump pompe (f) à vide de vapeur		паровой вакуумъ-насосъ (m) pompa (f) a vuoto a vapore bomba (f) de vacío á vapor	3
Eisenbahnwasserstation (f) railway pumping station station (f) de prise d'eau pour chemin de fer		желѣзнодорожная станція (f), имѣющая водоснабженіе; (часто:) желѣзнодорожное водоснабженіе (n); желѣзнодорожная водокачка (f) stazione (f) di rifornimento d'acqua per strade ferrate estación (f) de toma de agua ó de alimentación de agua para ferrocarriles	4

h) Gasdruckpumpen (f pl)

Gas Pressure Pumps (pl)

Pompes (f pl) à pression de gaz ou à gaz comprimé

Насосы (m pl) съ непосредственнымъ давленіемъ газа на жидкость

Pompe (f pl) a pressione di gas 5

Bombas (f pl) á presión de gas

Gasdruck (m) gas pressure pression (f) de gaz	давленіе (n) газа pressione (f) o tensione (f) del gas presión (f) del gas	6

H. A. HUMPHREY.
METHOD OF COMPRESSING ELASTIC FLUID.
APPLICATION FILED JULY 17, 1908. RENEWED JULY 14, 1917.

1,257,004. Patented Feb. 19, 1918.
4 SHEETS—SHEET 1.

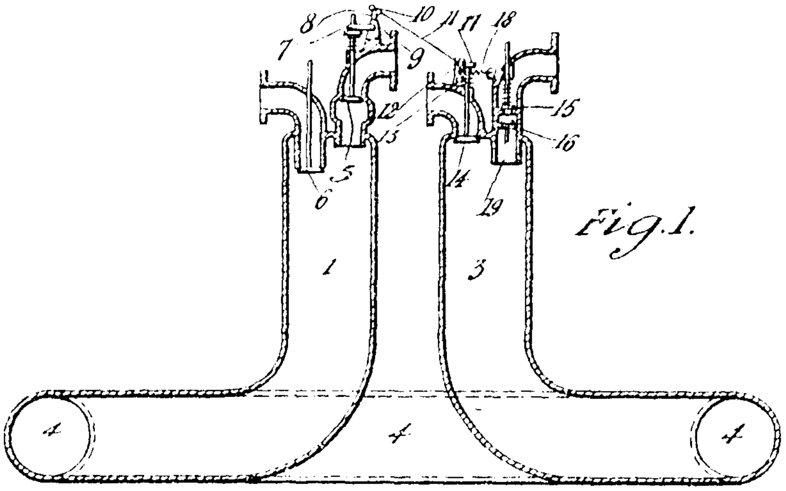


Fig. 1.

Fig. 2.

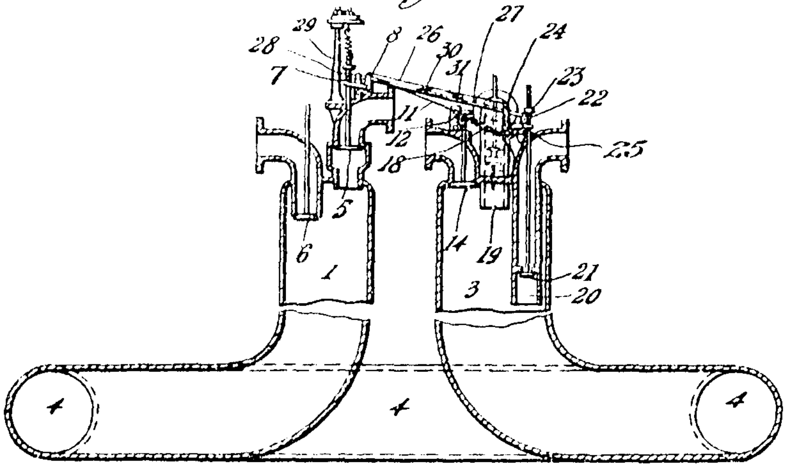
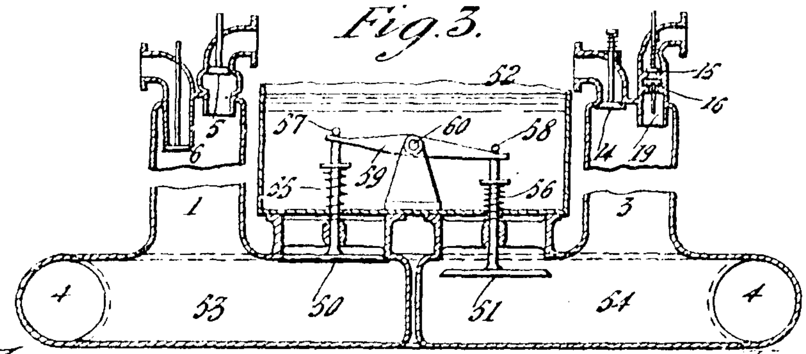


Fig. 3.



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H. A. HUMPHREY.

METHOD OF COMPRESSING ELASTIC FLUID.

APPLICATION FILED JULY 17, 1908. RENEWED JULY 14, 1917.

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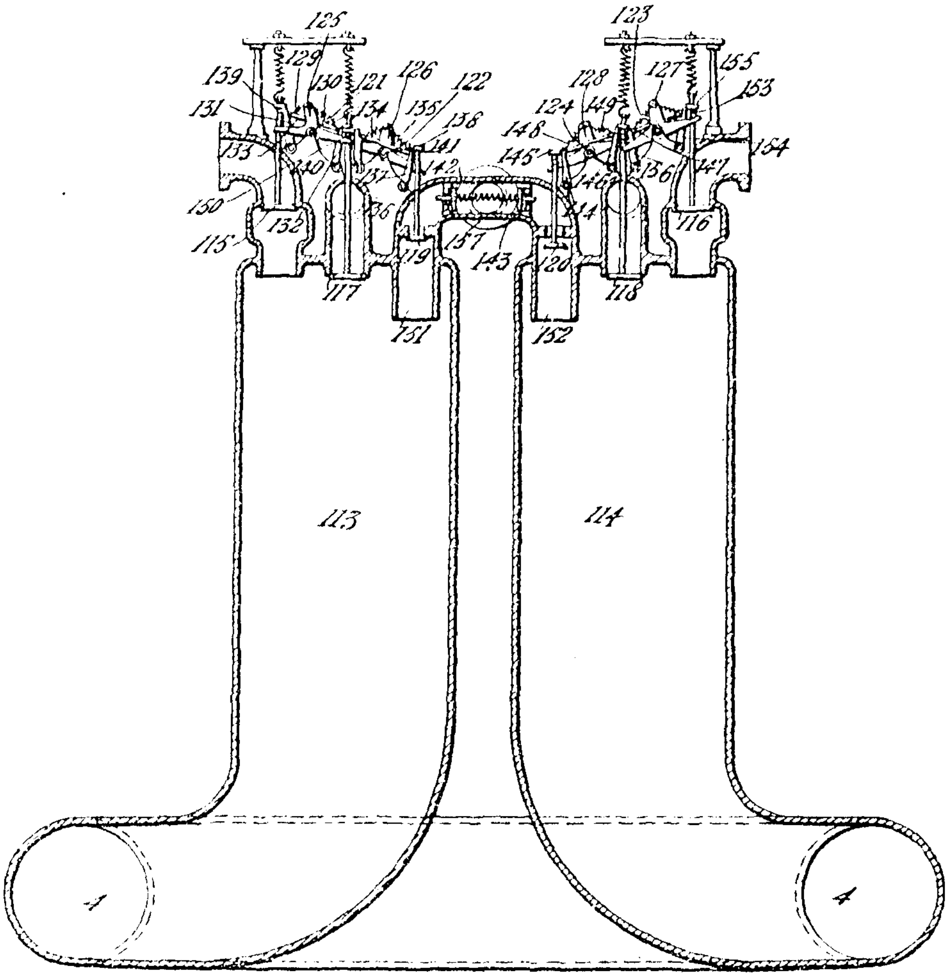
4 SHEETS—SHEET 2.

Fig. 1.



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4 SHEETS—SHEET 3.

Fig. 5.

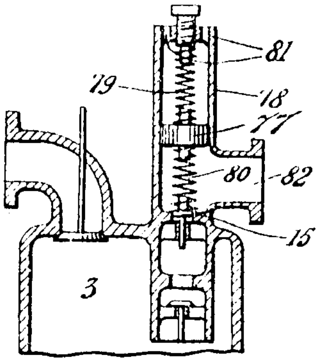


Fig. 6.

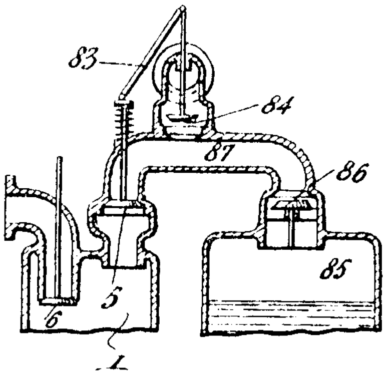
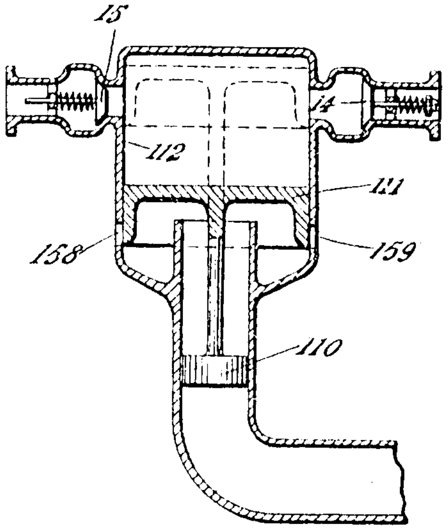


Fig. 7.



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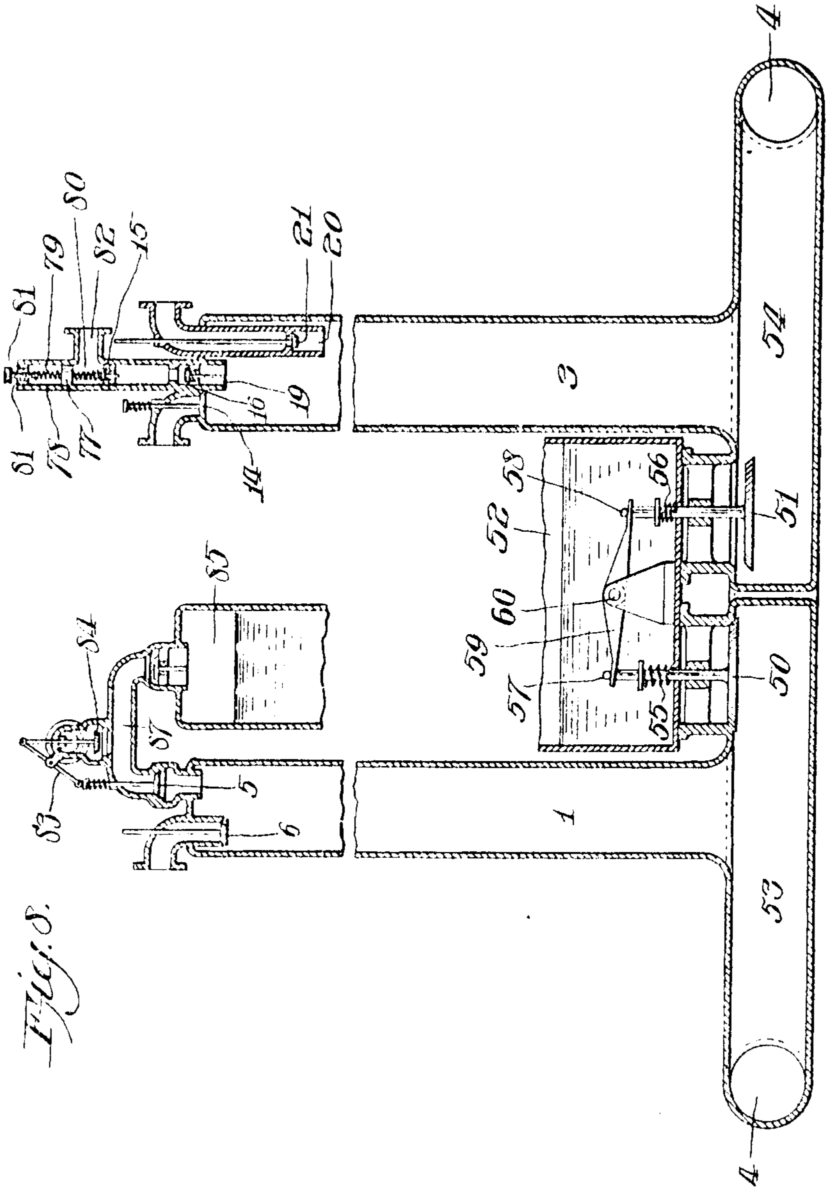
METHOD OF COMPRESSING ELASTIC FLUID.
APPLICATION FILED JULY 17, 1908. RENEWED JULY 14, 1917.

1,257,004. Patented Feb. 19, 1918.

4 SHEETS—SHEET 4.

Fig. 8.



INVENTOR
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By

ATTORNEY

WITNESSES
Max Hofmann
Howard

BY

UNITED STATES PATENT OFFICE.

HERBERT ALFRED HUMPHREY, OF WESTMINSTER, ENGLAND, ASSIGNOR TO HUMPHREY GAS PUMP COMPANY, A CORPORATION OF NEW YORK.

METHOD OF COMPRESSING ELASTIC FLUID.

1,257,004. Specification of Letters Patent. **Patented Feb. 19, 1918.**

Application filed July 17, 1908, Serial No. 444,061. Renewed July 14, 1917. Serial No. 180,682.

To all whom it may concern:

Be it known that I, HERBERT ALFRED HUMPHREY, a subject of the King of Great Britain, residing at 38 Victoria street, Westminster, in the county of London, England, consulting engineer, have invented certain new and useful Improvements Relating to Methods of Compressing Elastic Fluid, of which the following is a specification.

My invention relates to improvements in the method of utilizing heat energy in the movement of liquid and utilizing the momentum or kinetic energy of the moving liquid for compressing air or gases. The object of the present invention is to provide a method whereby the raising or forcing of liquid by an expansion of an ignited combustible charge is applied to the compression or blowing of air or gas. The cycle of operation is in many respects similar or analogous to that described and claimed in my copending applications Serial No. 438,425 filed June 13, 1908 and Serial No. 655,973 filed October 21, 1911, one of which is for a method and the other for an apparatus, which applications contain claims which may be read upon the disclosure of the present application.

Referring to the drawings which illustrate merely by way of example, suitable apparatus for effecting my invention—

Figure 1 is a diagrammatic vertical section of suitable apparatus for effecting my invention.

Fig. 2 is a similar view of a modified form of apparatus.

Fig. 3 is a similar view of another modified form.

Fig. 4 is a vertical section on a somewhat enlarged scale of another modified form of apparatus, wherein the combustion and compression chambers alternate in their functions.

Fig. 5 is a vertical section of a device for controlling the pressure at the air delivery.

Fig. 6 is a vertical section of a measuring device for combustible charges.

Fig. 7 is a vertical section of a device whereby the elastic fluid compressed, is separated from the compressing mass of liquid.

Fig. 8 is a vertical section of suitable apparatus for effecting my invention showing the coöperative relationship of the several features illustrated in the other figures.

Similar numerals refer to similar parts throughout the several views.

A form of the apparatus, which may be regarded as one of the simplest examples, inasmuch as it does not require that there shall be either an intake or discharge of fresh liquid at each cycle, is shown in Fig. 1, in which 1 is the combustion chamber and 3 is the compressor chamber, connected by a liquid duct or play pipe 4.

In the top of chamber 1 are fitted an admission valve 5 for combustible mixture, and an exhaust valve 6 for burnt products. Suitable valve gear is used so that the exhaust valve opens when the ignited combustible mixture has expanded to substantially atmospheric pressure or other desired pressure, and is shut when the rising liquid has exhausted sufficient of the products of combustion. The exhaust valve may conveniently be shut by impact of the rising liquid, and remain shut until the next ignited charge expands to atmospheric pressure. The admission valve opens by suction against the action of a light spring during the period at which it is desired to introduce the charge.

In the present example the stem of the admission valve carries a pin 7 capable of engaging with the bell crank 8 pivoted at 9 and carrying a pin 10 for engaging in the slotted end of rod 11 attached to a pawl 12 pivoted at 13 on the top of the compressor chamber.

At the upper end of the compressor chamber 3 is fitted an inlet valve 14 which opens against a light spring and is locked in an open position by pawl 12, which when pulled to the right by spring 18 engages above the collar 17 on the stem of air inlet valve 14, and so keeps this valve from closing again until this pawl is released by the motion of bell crank 8 which occurs when admission valve 5 opens. There is also fitted on the compressor chamber 3 a non-return outlet valve 15 controlled by a spring. In the outlet pipe there is also a valve 16 suitably supported and adapted to close on a seat just above it; this valve is so loaded, however, that the escaping air cannot close it, although the impact of the rising liquid is capable of doing so. The outlet pipe projects some distance into the compressor chamber, so that the liquid rising in the chamber can drive out air until the level of the liquid reaches the level of the projecting part 19 of the pipe, but further rise of the liquid imprisons a cushion of air in the top of the compressor chamber and causes liquid to be forced up the outlet pipe to shut valve 16, after which, all outlets being closed in the top of the compressor chamber, the imprisoned elastic cushion may be further compressed.

The method of operation may now be explained. The valves being in the position shown, it will be assumed that there is a compressed combustible mixture in the top of chamber 1 above the level of the liquid and that there is a charge of air or other elastic fluid in chamber 3 approximately at atmospheric pressure. Explosion occurs at or about maximum compression pressure. Any suitable mode of ignition may be employed. The increase of pressure thus produced drives the liquid downward and outward from chamber 1 through the pipe 4 which is long enough to permit a considerable portion of the energy to be transformed into momentum of kinetic energy of the moving liquid, and causes the liquid to rise in chamber 3, thus compressing the air therein. It may be assumed that on the discharge side of valve 15 there already exists compressed air at a pressure corresponding with that at which it is desired to deliver the air, so that valve 15 will not open until the pressure in chamber 3 exceeds somewhat the pressure at which the air is to be delivered, after which air will be discharged past valve 16 which remains open and past valve 15 which is opened under the action of the compressed air. According to the circumstances of the case, it may be that the level of the liquid in chamber 3 will not rise above the level of the lower projecting portion 19 of the inlet pipe, but if it does, a cushion is formed in the top of chamber 3 above this level, and valve 16 is shut by impact of liquid rising in the outlet pipe, in which case the further rise of pressure due to the compression of the elastic cushion will for the time being keep valve 16 shut.

Returning to the consideration of chamber 1, when expansion has reached atmospheric pressure valve 6 opens. On the column of liquid coming to rest, the compressed air in chamber 3 forces the column of liquid downward and outward from chamber 3, thus reversing the flow and causing liquid to rise in chamber 1. There being a free opening through the exhaust valve 6 to atmosphere, the liquid rising in chamber 1 expels the burnt products therefrom until, on attaining the level of valve 6 it shuts this valve by impact and compresses an elastic cushion in the top of chamber 1. In the meantime the energy of compression of the air contained in chamber 3 having been expended in imparting velocity to the column of water, the air arrives at atmospheric pressure, and the continued downward motion of the liquid in chamber 3 causes a fresh charge of air to be drawn in through valve 14, which is sucked open against the action of a light spring. When this occurs pawl 12, which previously rested against collar 17, is pulled by spring 18 and engages over the collar so that it locks the inlet valve in its open position. A point in the cycle is now reached where the column of liquid once more comes to rest and there is contained in the top of chamber 1 a compressed elastic cushion, in this case consisting of burnt products, or of burnt products and air, and in chamber 3 there is air at atmospheric pressure.

The compressed elastic cushion in the top of chamber 1 now expands, reverses the flow of liquid, and on the level of the liquid falling until atmospheric pressure is reached, which occurs when the level is about that of valve 6, the further motion of the liquid opens the admission valve 5 against the action of its light spring and draws in a fresh combustible charge. Exhaust valve 6 cannot open because it is locked closed in any suitable manner not shown. The admission valve 5 in opening operates bell crank 8, rod 11, and pawl 12, and so permits the spring of valve 14 to close the valve, thus stopping the discharge of air which was occurring as the liquid rose in chamber 3. This arrangement insures that the whole energy of expansion of the elastic cushion shall be utilized in giving velocity to the column of liquid until atmospheric pressure or thereabout is reached in chamber 1, after which admission valve 5 opens and valve 14 closes as described. The liquid continues to move from the combustion chamber end of the apparatus toward the compressor chamber, until its energy is expended in forcing liquid to rise in chamber 3 and compress the air therein, and then the liquid having once more been brought to rest, the compressed air in chamber 3 again reverses the direction of flow, causing liquid to again rise in chamber 1 and compress the fresh combustible charge therein, which, being ignited, commences a fresh cycle.

It should be mentioned that in this case bell crank 8, rod 11, and pawl 12 are provided for the better regulation of the working, but are not essential thereto, as the cushion expansion energy may suffice to draw in the necessary charge, although compressing air in chamber 3 throughout the cushion, expansion and charging stroke in chamber 1.

The amount of the charge admitted to chamber 1 is preferably measured by a suitable measuring device. An example of such a measuring device will be described later on.

The charge is measured not only to insure the regular working of the apparatus, but to enable the supply of combustible mixture to be cut off at the desired point so that the charge drawn in can be expanded below atmospheric pressure. In this case a partial vacuum is created in chamber 1 which assists the pressure in chamber 3 in imparting the necessary kinetic energy to the liquid to give the compression stroke in chamber 1.

In order to work the apparatus just described most advantageously, the pressure at which the air or gas is delivered should not exceed a certain limit, depending upon the mean effective area of the indicator diagram taken from the power cylinder, and consequently, when other ranges of pressure are required, the apparatus is modified as will be described.

In order that the operation of the compressor may be variable in the sense that it may deliver at each cycle either a large quantity of air at relatively low pressure or a small quantity of air at relatively high pressure, the modification shown in Fig. 2 may be adopted. Here, in addition to the parts already described and similarly numbered, chamber 3 has another pipe 20 communicating with the atmosphere and projecting for a considerable distance into the chamber. This pipe contains a valve 21 on the stem of which is a pin 22 and a collar 23 adapted to engage respectively with one arm of bell crank 24 and with pawl 25. The other arm of bell crank 24 is connected by rod 26 with bell crank 8, and pawl 25 is connected by rod 27 with another pawl 28, the latter being capable of engaging under collar 29, but the length of rod 27 is such that the two pawls cannot both at the same time be in engagement under their respective collars.

Two springs 30 and 31 have their ends attached to rods 26 and 27 respectively in such a manner that when rod 26 moves either to the right or the left rod 27 is pulled by the springs to follow it. There is also attached to bell crank 8 a spring, which is not shown, but acts so as to render stable the extreme positions of rod 26. Valve 21 is intended in this case to open under its own weight, its motion being limited by a spring or stop, and to be closed by the liquid rising in pipe 20 and impinging upon the valve. When admission valve 5 opens, pin 7, engaging with bell crank 8, moves rod 26 to the left where it remains until valve 21 opens and pin 22 engages bell crank 24 to move rod 26 to the right.

The action of the modified apparatus shown in Fig. 2 may now be described. The position of the valves as shown is correct for that part of the cycle where expansion of the ignited mixture has occurred in chamber 1, until the liquid, driven downward in chamber 1 and rising in chamber 3, has closed valve 21 which is locked shut by pawl 25. During the further expansion in chamber 1 the charge of air in chamber 3 is compressed and a portion of it is discharged under pressure into the air delivery pipe, but as before there remains a portion to form an elastic cushion. When the liquid comes to rest the energy of expansion of the cushion in chamber 3 causes the liquid to reverse its flow driving out products of combustion through exhaust valve 6, and, when this valve is shut by impact, giving the cushion stroke. When the level of liquid falling in chamber 3 is such that the pressure in chamber 3 arrives at about atmospheric pressure, valve 14 opens, admitting fresh air, and is locked open by pawl 12 under the action of spring 18; but valve 21 remains closed having been locked by pawl 25. Maximum cushion pressure having been attained in chamber 1 and the liquid again brought to rest, liquid is forced downward in chamber 1 and after reaching the pressure at which the admission valve opens a fresh charge is drawn in, pawl 28 being disengaged from under collar 29 so that the admission valve can open. The liquid rising in chamber 3 during the last mentioned portion of the cycle has no work of compression to do during the first part of its movement because valve 14 is open to the atmosphere, but when admission valve 5 opens and so operates bell crank 8 and rod 11, pawl 12 is pulled to the left and valve 14 shuts under the action of its spring, after which the motion of the liquid is arrested by compressing air in chamber 3 and it may be expanding the charge in chamber 1, below atmosphere. The charge having been drawn into chamber 1, and bell cranks 8 and 24, and rods 26 and 11 moved to the left, valve 5 shuts, pawl 28 engages under its collar, and pawl 25 is released from under the collar of valve 21, but as there is at this time pressure in chamber 3, valves 21 and 14 remain shut. There is now a compressed cushion of air in chamber 3 and a combustible charge in chamber 1, the pressure of which may be below atmosphere, and these conditions bring about a reversal of flow of the liquid which once more falls in chamber 3 and rises in chamber 1, compressing the charge in the latter. When atmospheric pressure or thereabout is reached in chamber 3 during this part of the cycle, valve 21 will open and, by its pin engaging with bell crank 24, will reverse the position of bell cranks 24 and 8 and rod 26 so that they are brought back to the position shown. The compressed charge in chamber 1 is now ignited, and liquid flows from that chamber toward the compressor chamber 3, expelling air from chamber 3 through valve 21 until the liquid arrives at the level of the bottom of the pipe 20, whereupon continued rise of the liquid shuts valve 21 and the cycle is once more brought back to the point from which the operations described are repeated.

From the foregoing description it will be evident that however much air may be drawn into chamber 3 the quantity which is compressed by the working stroke occurring in chamber 1 cannot exceed the capacity of the chamber above the level of the bottom of pipe 20, which level may be adjusted.

The action of the apparatus so far described may be varied to some extent by varying the amount of liquid contained therein; thus, reduction of the quantity of liquid is equivalent to increasing the capacity of the chambers, and increase of the quantity of liquid is equivalent to decreasing the capacity of the chambers.

Modifications of the apparatus will now be described, in which at each cycle fresh liquid is admitted and rejected. The combustion chamber and the compressor chamber are shown in Fig. 3 in such relative position that valves 50 and 51 for liquid, one near the combustion chamber and the other near the compressor chamber, may be conveniently placed close together with the object that their movements may be made to control one another. A tank 52 is so placed that there may be a free flow of liquid to the short pipes in which these valves are situated, which pipes are continued as at 53 and 54 to communicate with the respective chambers. As before, pipe 4 connects the combustion chamber and compressor chamber.

Assuming that chamber 1 is the combustion chamber, the exhaust valve will be that shown at 6, the admission valve being shown at 5. With all the valves in the position shown and a compressed combustible charge in the top of chamber 1 and a charge of air in chamber 3, ignition occurs in chamber 1 and liquid is forced downward and outward from chamber 1 toward chamber 3. Valve 51 is normally open so that liquid can escape past it into tank 52, while kinetic energy is being imparted to the moving column of liquid, when, however, the ignited gases in chamber 1 have expanded to a suitable extent the pressure below valve 50 is equal to that above this valve, and the further expansion of these gases causes valve 50 to open against the action of spring 55 thus admitting liquid to flow into the apparatus from tank 52. Valve 50 has on its stem a pin 57 adapted to engage against one arm of rocking lever 59 which is pivoted at 60 and the other arm of which engages against pin 58 on the stem of valve 51, so that the downward movement of valve 50 causes valve 51 to shut. Thus the energy of expansion of the ignited gases has been so far mainly transformed into kinetic energy of the moving column of liquid and when valve 51 shuts this kinetic energy is utilized in compressing the charge of air contained in chamber 3, some of which air is delivered under pressure past valve 15, but sufficient is retained to form an elastic cushion in which energy is stored for reversing the flow of the liquid. On reaching atmospheric pressure in the combustion chamber 1, exhaust valve 6 is opened under the action of its own weight and when the reverse flow occurs valve 50 is closed by the action of its spring and remains closed, owing to the pressure below the valve exceeding that above it. Liquid falls in chamber 3 and rises in chamber 1 expelling the exhaust products past exhaust valve 6 and on reaching this valve the liquid shuts it by impact, and compresses the elastic cushion contained in the top of the chamber. About the time when the pressure in chamber 3 reaches that of the atmosphere, valve 51 opens, also valve 14 is sucked open by the falling liquid and a fresh charge of air is drawn into chamber 3. Thus the level of liquid in chamber 3 tends to be maintained at the level of liquid in tank 52 and valve 14 shuts under the pressure of its own spring when the pressures are adjusted. Any further movement of liquid in pipe 4 draws in fresh liquid through valve 51.

The compressed elastic cushion in the top of chamber 1 now expands. Liquid is driven downward in chamber 1 along pipe 4 and escapes past valve 51 into tank 52 until, on the pressure falling sufficiently in chamber 1, fresh combustible mixture is drawn through valve 5 until the level of liquid in chamber 1 approaches that of the liquid in tank 52. The continued movement of the column of liquid in pipe 4 causes valve 50 to open, thus shutting valve 51. Fresh liquid is drawn in past valve 50 and the column of liquid proceeds to compress the air in chamber 3 until the work done brings it to rest, when valve 50 shuts. Valve 5 shuts under the action of its spring after the combustible charge has been drawn in, and valve 51 remains closed under the pressure now exerted upon it. The energy stored in the elastic cushion in chamber 3 is utilized to reverse the direction of flow, so that liquid falls in chamber 3, rises in chamber 1 and compresses the combustible charge therein, and toward the end of this flow of liquid valve 51 opens again, thus bringing all the parts once more into position for repeating the cycle.

From the foregoing description it will be seen that the working of the apparatus last described is suitable for compressing air to high pressures inasmuch as the energy of expansion of the ignited combustible gases is at first transformed into kinetic energy of the moving liquid and this energy may be made to operate upon a relatively small quantity of air and thus compress this air to a high pressure.

For compressing air to comparatively low pressures, the pressure of spring 55 on valve 50 is released so that this valve is normally opened under its own weight and the pressure of spring 56 is increased so that this valve is normally held shut. The effect of this change is that, when the working stroke occurs in chamber 1, valve 50 opens earlier, but as valve 51 remains closed; the liquid which is forced downward and outward from chamber 1 finds no outlet past valve 51, the compression of air starts from the commencement of the said working stroke, and a larger quantity of liquid rises in chamber 3 than in the case last considered. The energy of expansion of the ignited gases is therefore made to compress a larger volume of air, and that portion of the air which is delivered from the apparatus is delivered at a less pressure. As before, a portion of the air is retained in chamber 3 to supply the energy to reverse the flow of liquid.

After reversal and the expansion of the air cushion in chamber 3, air is drawn into this chamber until the level of the liquid approximates to that of the liquid in tank 52, liquid meanwhile escaping past valve 50 after rising in chamber 1. The level of liquid in chamber 1 will at this time be approximately the level of the liquid in tank 52, some of the products being already exhausted. The continued movement of the column of liquid in pipe 4 now causes valve 51 to open, thus closing valve 50, so that the liquid is forced to rise in chamber 1 to complete the exhaust and give the usual cushion stroke. The liquid, having come to rest, its flow is reversed under the action of the energy of the cushion in chamber 1, and the cushion expansion stroke, followed by the intake of a fresh combustible charge in this chamber, during which air in chamber 3 is compressed. Before the movement of the liquid ceases, valve 50 will open, while valve 51 remains shut. The liquid comes to rest, its flow is again reversed due to the pressure in chamber 3, and liquid flows downward in this chamber while the compressed air is expanding. The liquid which would otherwise rise in chamber 1 is at this time escaping through valve 50 into the tank, while the energy of expansion of the air in chamber 3 is being transformed into kinetic energy of the moving liquid. When, however, the point in this expansion is reached at which valve 51 opens, thus shutting valve 50, the kinetic energy of the liquid is utilized in compressing the combustible charge in chamber 1. When this kinetic energy is expended the liquid comes to rest and everything is ready for the starting of a fresh cycle by the ignition of the charge in chamber 1.

It will be seen that more liquid has entered and left chamber 3 than has entered and left chamber 1, and that the volume of air operated upon in chamber 3 is greater than in the previous case, so that the pressure at which the compressed air is delivered will be correspondingly less.

For some purposes it is allowable that the compressed air delivered from the apparatus should be mixed with products of combustion, as for instance, when the compressed air is to be used for driving a motor, and when this is the case the pump and the compressor may be interchangeable in respect of their functions, so that combustion occurs in each alternately. Such an arrangement is shown in Fig. 4, where 113 and 114 are the chambers which serve alternately as combustion chambers and air compressor chambers.

In the top of chamber 113 are fitted an inlet valve 115 for combustible mixture, an inlet valve 117 for air, and an exhaust valve 119 for the delivery of a mixture of compressed air and burnt products. Corresponding parts 116, 118 and 120 are fitted in the top of chamber 114. Rigidly attached to the top casting of chamber 113 are pivots 121 and 122 about which move two 3-arm cranks 125 and 126. The left and right hand arms are adapted to engage with pins fitted to the stems of valves 115, 117, and 119 as shown in said Fig. 4. The top arm of crank 125 is attached by springs 129 and 130 to pawls 131 and 132 respectively, and these pawls are connected by rod 133. The top arm of crank 126 is connected by springs 134 and 135 with pawls 136 and 137 respectively which are connected by rod 138. Pawl 131 is adapted to engage under a collar 139 on the stem of valve 115. Pawls 132 and 136 are adapted to engage under the collar 140 on the stem of valve 117 and pawl 137 is adapted to engage under the collar 141 on the stem of valve 119. In the outlet pipe there is, in addition to valve 119, a spring-controlled valve 142. Corresponding parts are fitted to the top of chamber 114.

The action of the apparatus is as follows:—The illustrated position of the valves is correct for that part of the cycle in which there is a compressed combustible charge in the top of chamber 113 and a charge of air and burnt products in chamber 114. Ignition now occurs in chamber 113 and the liquid is forced downward and outward from this chamber and rises in chamber 114 expelling the mixed charge at the desired pressure past valves 120 and 143 into the outlet pipe 157. After the liquid has attained the level of the inwardly extending pipe 152 it rises in this pipe and shuts valve 120 by impact. This valve is at once locked by pawl 144 engaging under collar 145 and simultaneously pawl 146 is released from under collar 147, this motion being obtained by the tension on spring 149 which, in the position shown, is greater than the tension on spring 148. The further rise of the liquid in chamber 114 compresses the elastic cushion in the top thereof and stores the energy to give the reverse flow. When the combustible products in chamber 113 have expanded to atmospheric pressure, valve 117 is sucked open and air is drawn in to mix with the products of combustion and this motion of the valve moves cranks 125 and 126 through pin 150 engaging with them. When the suction stroke in chamber 113 is finished and the liquid comes to rest, valve 117 closes under the action of its spring and pawls 132 and 136 engage under its collar and pawls 131 and 137 are disengaged from under collars 139 and 141 respectively, this movement being brought about by the fact that in the new position of cranks 125 and 126 the tension of springs 129 and 135 is increased and that of springs 130 and 134 diminished. Valve 119 being now released falls by its own weight, and when the energy stored in the elastic cushion in the top of chamber 114 causes the reverse flow, the liquid rising in chamber 113 has a large volume to compress and will not reach the level of the bottom of pipe 151, nothing happens therefore in this chamber except the compression of the mixture therein. But in chamber 111 when the pressure has fallen to atmosphere the further movement draws in a combustible charge past valve 116 and the movement of this valve, causing pin 153 to engage against crank 127, changes the position of this crank and when the valve shuts again under action of its spring, pawl 154 engages under collar 155 and locks the valve, while pawl 156 is withdrawn from under collar 147. The liquid having again come to rest and the pressure in chamber 113 being higher than in chamber 114, a second outward flow occurs from chamber 113 and the combustible charge is compressed in chamber 111 and the ignition of this charge starts a fresh cycle, in which the functions of the chambers are reversed. Springs, not shown, are used to render stable the extreme positions of cranks 125, 126, 127 and 128, in the manner already referred to for other similar cases. Preferably a measured charge of combustible mixture is used.

It has been assumed, in describing the cycles of the various modifications of apparatus, that there already existed a pressure in the outlet pipe of the compressor chamber on the discharge side of the outlet valve equal to that at which it was desired to deliver the compressed air. This may, however, not be the case in starting the apparatus, as for instance, when a reservoir for containing the compressed air delivered has to be pumped up to the desired pressure. If the compressor started without pressure in the delivery pipe the irregular working, which would otherwise result until the desired pressure had been attained, may be avoided by fitting to the outlet pipe the simple device shown in Fig. 5. A movable piston 77 is held in position in a cylinder 78 by two springs 79 and 80. On the top side of the piston air is maintained at atmospheric pressure by holes 81 which communicate with the atmosphere, while the under side of the piston is subject to the pressure of the air in the delivery pipe 82. The piston is shown in the position which it assumes when there is atmospheric pressure in the pipe 82. This position, due to the pressure of spring 79, is the piston's lowest position and spring 80 is compressed to such an extent that it requires a pressure below the outlet valve 15 equivalent to the desired pressure at which air is to be delivered, to open this valve. As the air compressor is set to work and air is discharged past valve 15 the pressure in pipe 82 begins to rise and the pressure on the under side of the piston 77 gradually increases until the desired working pressure for the compressed air is reached, when piston 77 has been moved upward against the pressure of spring 79. This upward movement relieves the pressure on spring 80 and the springs may be so adjusted that little or no compression now remains in spring 80. Consequently valve 15 now opens when the pressure on the under side thereof is substantially equal to the pressure at which the air is to be delivered. By this arrangement the pressure at which valve 15 opens may be maintained practically constant from the starting of the apparatus until the desired pressure of delivery is attained, and thus the pressure conditions for each cycle remain the same, in spite of the varying pressures at the delivery side of the outlet valve.

A device for measuring a combustible charge into the combustion chamber has already been alluded to, and a description of such a device will now be given in connection with Fig. 6.

The combustion chamber 1 is fitted with the usual inlet valve 5, but the stem of this valve has a connection, shown diagrammatically as a rod 83, with the stem of valve 84 placed in the combustible inlet pipe, so that when valve 5 opens valve 84 is closed. The combustible inlet pipe 87 is also connected with a measuring chamber 85, in the top of which there is a valve 86 adapted to be shut by impact of liquid upon it. Measuring chamber 85 is connected with a supply of water not shown which is open to the atmosphere in such a manner, that the normal level of water in the measuring chamber may be adjusted.

The action of the apparatus is as follows:—

When valve 5 opens, valve 84 is closed and the suction in chamber 85 causes the liquid to rise therein until valve 86 shuts by the impact of liquid upon it. Thus, valves 84 and

86 being both closed, no more combustible mixture can be taken into chamber 1, and when the pressure in this chamber and in the combustible inlet pipe 87 has risen sufficiently, valve 5 will shut under the action of its spring. This movement of valve 5 opens valve 84, and further combustible mixture flows into pipe 87. Valve 86 falls by its own weight, and the liquid in chamber 85, in falling to its normal level, draws a measured quantity of combustible mixture into the chamber.

The seats of valves 5 and 84 are prolonged in such a manner as to cause these valves to act partly as piston valves, so that valve 84 may be shut by the time valve 5 is open.

It will be seen that by altering the normal level of liquid in chamber 85, the quantity of the measured charge taken in can be varied.

It is well known that pistons of different areas may be used to convert high pressure energy into low pressure energy, or vice versa, and it is obvious that this fact can be utilized in connection with the present invention. Thus, if low pressure air in large volume is required, the liquid, instead of rising and falling in an air compressor chamber, may act directly upon a piston 110 (Fig. 7) fitted in a turned portion of the pipe in which the liquid moves. This small piston may be connected with a larger piston 111 and the latter may move in an air compressor chamber 112. 14 and 15 are inlet and outlet valves respectively fitted in the chamber and so situated that piston 111 in moving to its top position closes both inlet and outlet and imprisons a cushion of air in the top of the chamber. Similarly, when the piston moves to its bottom position, it closes the vents to the atmosphere 158 and 159 and so cushions the air imprisoned in the bottom of the chamber and prevents the piston striking the chamber. The action of the apparatus is precisely on the same lines as when liquid rises and falls in the chamber, the only difference being that, instead of the liquid rising and shutting valve 16 in chamber 3, to imprison an air cushion which stores the energy for reversing the flow of liquid, piston 111 cuts off communication by covering the inlet and outlet pipes in chamber 112 and thus imprisons an air cushion in the top of this chamber. The air cushion in the bottom of the chamber is not intended normally to come into play as the lowest position to which the piston normally reaches is that shown in Fig. 7.

Throughout the present specification the elastic cushion which is compressed in the air compressor to furnish the energy for the return flow of liquid has been spoken of as a cushion of air, but it has been shown in my application No. 438,126 filed Feb. 12, 1910, that in the case of the pump the elastic cushion may be obtained by a piston or valve in a branch chamber opening into the combustion chamber, that side of the piston which is not presented to the chamber being acted upon by a spring or by compressed air or equivalent device. It is obvious that a similar method may be employed in connection with the present invention without departing from the spirit thereof.

What I claim is:—

1. The method of compressing air or other elastic fluid which consists in propelling a mass of liquid outwardly from a combustion chamber by the energy of expansion of an ignited combustible charge, allowing the said outwardly propelled mass to compress elastic fluid in a compression chamber, allowing some of the energy of this compression to cause the said mass to move inwardly to compress an elastic cushion in the combustion chamber and to cause an intake of elastic fluid in the compression chamber, allowing the energy of the compression of the cushion to cause the mass to flow outwardly again to entrain a fresh combustible charge and compress the admitted elastic fluid, and allowing the energy of the compression of the said admitted fluid to cause the mass to flow toward the combustion chamber to compress the fresh charge.

2. The method of compressing air or other elastic fluid, which consists in propelling a mass of liquid outwardly from a combustion chamber by the energy of expansion of an ignited combustible charge, allowing the said outwardly propelled mass, while a portion of the liquid is raised to a height, to compress elastic fluid in a compression chamber, allowing some of the energy of this compression, aided by the liquid which has been elevated, to cause the said mass to move inwardly to compress an elastic cushion in the combustion chamber, and to cause an intake of elastic fluid in the compression chamber, allowing the energy of the compression of the cushion to cause the mass to flow outwardly again to entrain a fresh combustible charge and compress the admitted elastic fluid, and allowing the energy of the compression of the said admitted fluid to cause the mass to return toward the combustion chamber to compress the fresh combustible charge.

3. The method of compressing air or other elastic fluid which consists in propelling a mass of liquid outwardly from a combustion chamber to compress elastic fluid in a compression chamber, allowing some of the energy of compression to cause the said mass to move inwardly to compress an elastic cushion in the combustion chamber and to cause an intake of elastic fluid in the compression chamber, allowing the energy of the compression of the cushion to cause

the mass to flow outwardly again to entrain a fresh combustible charge and compress the admitted elastic fluid allowing the energy of the compression of the said admitted fluid to cause the mass to flow toward the combustion chamber to compress the fresh combustible charge, and varying the volume of elastic fluid compressed by varying the volume of liquid that is forced into the compression space in the said compression chamber by the said energy of expansion.

4. The method of compressing air or other elastic fluid which consists in propelling a mass of liquid outwardly from a combustion chamber to compress elastic fluid in a compression chamber connected with the combustion chamber, allowing the energy of compression to cause the said mass to move inwardly to compress an elastic cushion in the combustion chamber and to cause an intake of elastic fluid in the compression chamber, allowing the energy of the compression of the cushion to cause the mass to flow outwardly again to entrain a fresh combustible charge and compress the admitted elastic fluid, allowing the energy of the compression of the admitted fluid to cause the mass to flow toward the combustion chamber to compress the fresh combustible charge, and varying the volume of the air or elastic fluid compressed by allowing liquid to be taken in and rejected during a cycle.

5. The method of compressing air or other elastic fluid which consists in propelling a mass of liquid outwardly from a combustion chamber to compress elastic fluid in a compression chamber connected with the combustion chamber, allowing some of the energy of compression to cause the said mass to move inwardly to compress an elastic cushion in the combustion chamber and to cause an intake of elastic fluid in the compression chamber, allowing the energy of the compression of the cushion to cause the mass to flow outwardly again to entrain a fresh combustible charge and compress the admitted elastic fluid, allowing the energy of the compression of the admitted fluid to cause the mass to flow toward the combustion chamber to compress the fresh combustible charge, and varying the volume of the elastic fluid compressed by allowing some of the elastic fluid taken in to be rejected again previous to compression.

6. The method which consists in reciprocating a column of liquid, with a velocity sufficiently limited to preserve the coherence of the column and having sufficient bulk and path of travel to acquire useful momentum, by applying the actuating force of an expansible medium to each end of the column at required intervals respectively, and utilizing the pressure of the liquid column due to the momentum it acquires in its reciprocating movements.

7. The method which consists in reciprocating a column of liquid with a velocity sufficiently limited to preserve the coherence of the column and having sufficient bulk and path of travel to acquire useful momentum by applying the actuating force of an expansible medium to each end of the column at required intervals respectively and utilizing said momentum.

8. The method which consists in reciprocating a column of liquid with a velocity sufficiently limited to preserve the coherence of the column and having sufficient bulk and path of travel to acquire useful momentum by applying the actuating force of an expansible medium to each end of the column at required intervals respectively and utilizing said momentum to compress an elastic medium.

9. The method of compressing elastic fluid which consists in reciprocating a body of liquid, the first outward movement of said reciprocation being due to the expansive force of an ignited combustible charge, utilizing the momentum of said outward movement to compress the elastic fluid and to deliver a portion thereof, utilizing the expansion of the remaining compressed fluid to cause a return movement of the liquid and automatically regulating the pressure of compressed fluid at the discharge, during the earlier cycles until normal pressure conditions are established.

10. The method which consists in reciprocating a column of liquid with a velocity sufficiently limited to preserve the coherence of the column and having sufficient volume and path of travel to acquire useful momentum, one outstroke movement of said reciprocation due to an expansive force utilizing said momentum of said outstroke to compress an elastic fluid and deliver a portion thereof.

11. The method which consists in reciprocating a column of liquid with a velocity sufficiently limited to preserve the coherence of the column and having sufficient volume and path of travel to acquire useful momentum, one outstroke movement of said reciprocation due to an expansive force, utilizing the momentum of said outstroke to compress an elastic fluid and deliver a portion thereof, and utilizing the energy of another portion of said compressed elastic fluid in causing a return movement of the liquid.

12. The method which consists in reciprocating a column of liquid of sufficient volume and path of travel to acquire useful momentum, one movement of said reciprocation due to an expansive force, utilizing said momentum to compress an elastic fluid and deliver a portion thereof and varying the amount of elastic fluid compressed.

13. The method which consists in reciprocating a column of liquid of sufficient volume and path of travel to acquire useful momentum, one movement of said reciprocation due to an expansive force, utilizing said momentum to compress an elastic fluid and deliver a portion thereof and varying the amount of elastic fluid compressed while the expenditure of initial energy remains constant.

14. The method of compressing elastic fluid which consists in reciprocating a body of liquid, the first outward movement of said reciprocation being due to the expansive force of an ignited combustible charge, utilizing the momentum of said outward movement to compress the elastic fluid and to deliver a portion thereof, utilizing the expansion of the remaining compressed fluid to cause a return movement of the liquid, and introducing and discharging by regulatable means liquid for controlling the volume of compressed elastic fluid discharged.

15. The method of compressing elastic fluid which consists in reciprocating a body of liquid, the first outward movement of said reciprocation being due to the expansive force of an ignited combustible charge, utilizing the momentum of said outward movement to compress the elastic fluid and to deliver a portion thereof, utilizing the expansion of the remaining compressed fluid to cause a return movement of the liquid, and introducing and discharging by regulatable means liquid for controlling the pressure of compressed elastic fluid discharged.

16. The method of compressing elastic fluid which consists in reciprocating a body of liquid, the first outward movement of said reciprocation being due to an expansive force, utilizing the momentum of said outward movement to compress the elastic fluid and to deliver a portion thereof, utilizing the expansion of the remaining compressed fluid to cause a return movement of the liquid and introducing and permitting the escape by regulatable means of fresh elastic fluid for controlling the pressure or the volume of said fluid compressed and delivered.

17. The method of compressing elastic fluid which consists in reciprocating a body of liquid, the first outward movement of said reciprocation being due to the expansive force of an ignited combustible charge, utilizing the momentum of said outward movement to compress the elastic fluid and to deliver a portion thereof, utilizing the expansion of the remaining compressed fluid to cause a return movement of the liquid and utilizing measuring means for securing the introduction into the combustion chamber at each cycle of a definite measured combustible charge.

18. The method which consists in reciprocating a column of liquid of sufficient volume and path of travel to acquire useful momentum, one movement of said reciprocation due to an expansive force, utilizing said momentum to compress an elastic fluid and deliver a portion thereof, and varying the pressure of compression of said elastic fluid.

19. The method which consists in reciprocating a column with a velocity sufficiently limited to preserve the coherence of the column and having liquid of sufficient volume and path of travel to acquire useful momentum, one movement of said reciprocation due to an expansive force, utilizing said momentum to compress an elastic fluid and deliver a portion thereof and definitely measuring the expansible charge used in giving the initial movement of reciprocation.

20. The method which consists in reciprocating a column of liquid of sufficient volume and path of travel to acquire useful momentum, one movement of said reciprocation due to an expansive force, utilizing said momentum to compress an elastic fluid and deliver a portion thereof, measuring the expansible charge used in giving the initial movement of reciprocation and causing a partial vacuum at the rear of the outstroke introducing said charge to facilitate the return stroke.

21. The method which consists in reciprocating a column of liquid of sufficient volume and path of travel to acquire useful momentum, one movement of said reciprocation due to an expansive force, utilizing said momentum to compress and discharge an elastic fluid, and maintaining a constant pressure at the discharge.

22. The method which consists in reciprocating a column of liquid with a velocity sufficiently limited to preserve the coherence of the column and having sufficient volume and path of travel to acquire useful momentum, one movement of said reciprocation due to an expansive force, utilizing said momentum to compress and discharge an elastic fluid and maintaining automatically a constant pressure at the discharge.

In testimony whereof I have signed my name to this specification in the presence of two subscribing witnesses.

HERBERT ALFRED HUMPHREY.

Witnesses:
WALTER J. SKERTEN,
JOSEPH MILLARD.

Copies of this patent may be obtained for five cents each, by addressing the "Commissioner of Patents, Washington, D. C."

the pressure, the volumes (as measured under the varying pressure) are about constant.

The figures given in the absorption table are for atmospheric oxygen and nitrogen, not single, separate gases. They therefore represent conditions just as one would find them in hydraulic compression. The table shows that water absorbs from 80 to 90 per cent. more of nitrogen from the atmosphere than of oxygen.

OXYGEN ABSORBED MORE READILY THAN NITROGEN

If, however, these tabular percentages be divided by 0.21 for the oxygen and by 0.79 for the nitrogen, being the proportions of each gas in 1 cu. ft. of air, it is seen that for the same volume of gas offered for absorption, oxygen is absorbed more than twice as fast as nitrogen, and the composition of the air taken up by the water is richer in oxygen than the original air. As the absorbed air is mostly carried away by the water flowing from the uptake shaft, the air which remains and which is used for industrial purposes is poorer in oxygen than the original air. The absorbed air is largely given up from the water, but only after the water has risen in the uptake till the pressure is reduced, and even then is largely held in minute globules which give the water a milky appearance.

The Humphrey Pump and Compressor.—The changes which are occurring in the means and methods of power development are having their effect upon the work of air compression and wider departures from established practice are likely to follow. While at the beginning the larger air-compressors were quite generally steam-driven, many now are operated by gas- or oil-engines, saying nothing here of direct and indirect drives by water-power and by electricity however generated. The oil-engine working mechanically, with reciprocating pistons and rotating shafts, is finding a rival for pumping service in the Humphrey pump which dispenses with the mechanical devices and yet shows the highest efficiencies, using completely all the expansive force of the successive charges of explosive mixture. Then, as an afterthought or a following up of the invention, the column of water set in motion by the explosion, instead of having the further end of it driven into a water delivery pipe, is made to expend its force in compressing in a suitable chamber, and then expelling into the compressed-air system, a volume of air commensurate with the disposable force.

It is not easy to tell the story of the Humphrey pump in few words. The following is mostly abstracted from a paper by its inventor, Mr. H. A. Humphrey, before the Manchester Association of Engineers:

The simplest form of Humphrey pump is shown in Fig. 39. Imagine a charge of gas and air to be compressed in the top of chamber C and fired by a sparking plug projecting through the top casting. All valves are closed when the explosion occurs and the increase in pressure drives the water downward in C, setting the whole column of water in the discharge pipe D in motion. The column attains kinetic energy while work is being done on it by the expanding gases, and may move with considerable velocity when these have reached atmospheric pressure. The motion of the water column cannot be suddenly arrested, hence the pressure in the combustion chamber C tends to fall below that of the atmosphere, the exhaust valve E opens, and also the water valves VV in the supply tank ST. Water rushes in through VV mostly to follow the moving column in pipe D, but partly to rise in C in an effort to reach the same level inside the chamber as exists in ST.

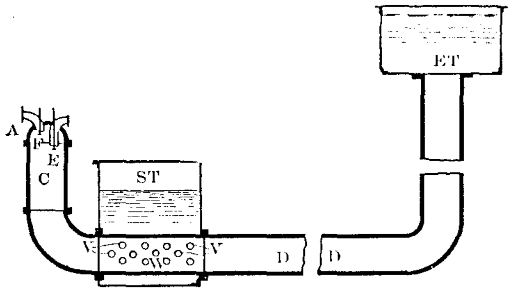


Fig. 39.—Simplest Type of Humphrey Pump.

When the kinetic energy of the moving column has expended itself by forcing water into the high-level tank ET it comes to rest, and there being nothing to prevent a return flow, the column starts to move back toward the pump, and gains velocity in the return direction until the water reaches the level of the exhaust valve E, which it shuts by impact. A certain quantity of burnt products is now imprisoned in the cushion space F, and the energy of the moving column is expended in compressing this gas cushion to a greater pressure than that due to the static head of the water in tank ET. Hence a second outward movement of the column results, and when the water reaches the level of valve E the pressure of the space F is again atmospheric, and further movement of the water opens valve A against a light spring, and draws in a fresh charge of gas and air. If there were no friction, the water would fall to the same level as that from which the last upward motion started, but the amount of combustible charge drawn in is slightly less than this movement would represent. Once more the column of water returns under the elevated tank pressure, and compresses the charge of gas and air, which is then ignited to start a fresh cycle of operations.

The action of the pump is not altered if, instead of delivering into an elevated tank, it discharges into an air vessel, or into an open-top standpipe or tower, and both these arrangements are useful if a continuous flow from the outlet is desired.

In the simple form of pump the degree of compression of the combustible charge prior to ignition depends on the height to which the water is raised, and exceeds the static equivalent of the head. This is obvious if one remembers that the kinetic energy acquired by the liquid column on its return flow is utilized in compressing the combustible gas, while the compression brings the column to rest.

The same considerations enter into the question of the cushion pressure attained, but here we are dealing with the compression of a volume of gaseous fluid which occupies the clearance-space only, and the stroke of the water column is greater in proportion, because for the first part of the stroke exhaust products are being expelled, and no compression occurs. The cushion pressure rises rapidly as the height to which the water is lifted increases, and at the maximum lift of about 40 ft. to which the simplest type of pump is limited, it may exceed the explosion pressure when using producer-gas, or may approximately equal the explosion pressure when working with city gas or gasoline.

A Humphrey pump was exhibited in operation at the Brussels International Exhibition, and obtained two "highest possible" awards—namely, a Grand Prix in the class for gas-engines, and a Grand Prix in the class for pumps. Its construction is similar to that in Fig. 39, and explanation is not needed till we come to the valve gear shown in Figs. 40 and 41. It will be observed that a bolt B sliding horizontally must lock either the admission-valve A or the exhaust-valve E by engaging under collars a or e, which are fixed on the stems of their respective valves. Now the bolt is urged right or left, according to whether spring s_1 or s_2 is pulling the hardest, and this again

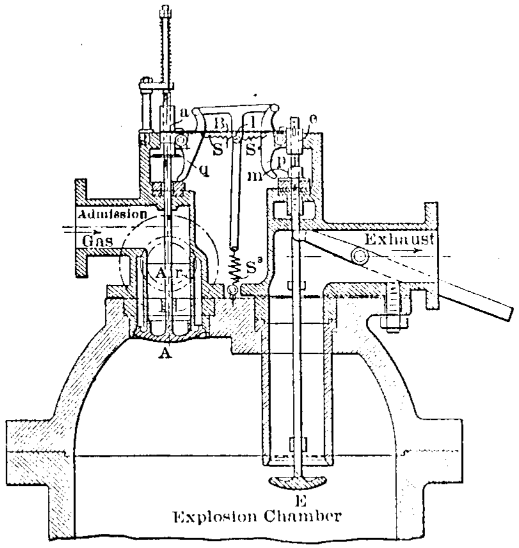


Fig. 40.

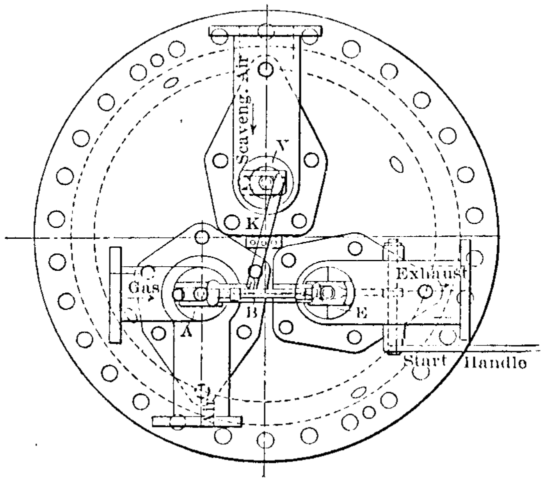


Fig. 41.—Vertical Section and Plan Showing Valve Gear of Humphrey Pump.

depends on whether the link l, to which the springs are attached, has been shifted to the right or left. Suppose the exhaust-valve opened last, then its washer m, engaging against cam arm *p*, moves the system p, l, q, so that it leans to the right, in which position it is retained by the tension of spring s_3. This puts tension on spring s_1 and loosens spring s_2; bolt *B*, therefore, tries to move to the right, but until the exhaust valve shuts it can only press upon collar *e*. However, when valve *E* comes on its seat the bolt instantly locks under *e*, and the same motion which holds valve *E* shut has released valve *A*, so that next time a suction occurs in the combustion-chamber, *A* only can open. Precisely the same kind of action occurs when *A* shuts and is locked and *E* is released again. Thus valves *A* and *E* are automatically allowed to act alternately, the difference between them being that while *E* remains open till shut by the rising water, *A* shuts under the action of its supporting spring, so soon as the suction in the chamber permits the spring to lift the valve to its seat.

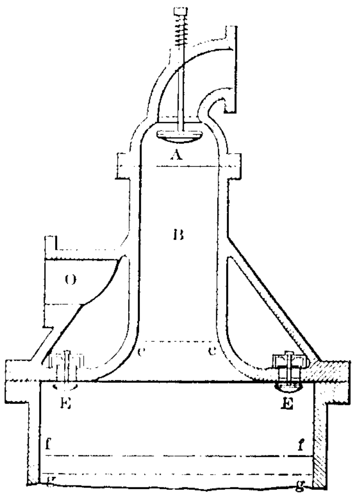


Fig. 42.

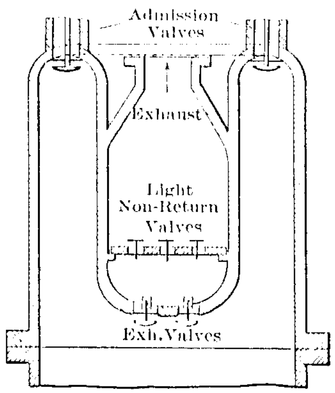


Fig. 43.

Figs. 42-43.—Alternative Arrangements of Two-cycle Pump Without Valve Gear.

The scavenging-valve *V* is shown in the plan of the combustion-head, Fig. 41, and as it operates at the end of each expansion stroke, its locking and release periods correspond with those of the exhaust-valve, and are made simultaneous by a lever pivoted at *K*, and operated by a pin on the bolt *B*. If the water could rush in fast enough when the pressure falls to atmosphere, there would be no scavenging action; but the incoming water has to be accelerated, and that just gives rise to a sufficient suction to effect the desired scavenging. In the exhaust outlet there is a light non-return valve, to prevent burnt products being drawn back into the chamber.

Figs. 42 and 43 show alternative arrangements of the top of the combustion chamber for a two-cycle pump requiring no valve gear. The combustion chamber has to be specially shaped, so that the incoming charge, which may be preceded by pure air, displaces the burnt products and mixes as little as possible with them. Thus, in Fig. 42, *A* is the admission valve at the top of the tall, narrow part of the chamber *B*, in which the full charge volume extends down to the level *cc*. *A* number of exhaust valves *E* lead to a common exhaust outlet *O*, which may be fitted with a non-return valve, or each exhaust valve may carry a light non-return valve on its spindle, as shown. The level at which expansion reaches atmospheric pressures is, say, *ff*, but this level having been reached by the water, its further movement draws in fresh combustible mixture till it occupies the space down to *cc*, and the liquid level has fallen to *gg*. The column of liquid now returns and drives the exhaust products through the valves *E*—which had opened by their own weight—until these valves are shut by the water. The kinetic energy acquired by the column is now spent in compressing the fresh charge, which is ignited to start a new cycle. Thus, each outstroke is a working stroke, and no locking gear is required on the valves.

The same cycle applies to Fig. 43, but in this case there is a series of admission valves placed in a ring so as to allow the mixture to enter with a low velocity in order to prevent eddies and mixing with the exhaust products. A higher compression pressure is obtained with this pump than with the simple pump, and consequently higher efficiencies with the same lift.

Fig. 44 shows the arrangement of a double-barrel pump, which has two combustion chambers, *A* and *B*, in which explosion occurs alternately. Any Humphrey pump, whether single or double barrel, may be converted into a high-lift pump by means of an air-vessel fitted with valves, and called an "intensifier." The idea is to first allow the water-column to gain velocity, and then to utilize its kinetic energy to (a) compress an elastic fluid, and (b) deliver water under the pressure to which the elastic fluid has been compressed.

In Fig. 44, *A* and *B* are the barrels of a two-barrel pump, and at the end of the play-pipe *D* there are two air-vessels *E* and *F*, the latter being large enough to give a continuous flow at outlet *O*, and to maintain a practically uniform pressure. The smaller air-vessel *E* is fitted with a downwardly projecting pipe *K*, open to the atmosphere at the top, and carrying a valve *L* at its lower extremity, arranged to close under the action of the rising water. The cycle starts with explosion, all valves except *L* being shut, and the water level as shown. While the water level in *E* is rising to *L*, air is merely being discharged into the atmosphere, and as no work is being done by the column of water, it gains speed until valve *L* is shut by impact. Imprisoned in *E* there is now a definite quantity of air, which suffers compression until its pressure reaches that at which the high-pressure water-valves *W* can open, and allow the remaining kinetic energy of the column to force water into *F*. Valves *W* close when the column comes to rest, but there remains enough energy in the compressed air in *E* to give, by expansion, the return flow, which causes exhaust in *A* and compression of the fresh charge in *B* to start a fresh cycle. When the water level falls below valve *L* this valve opens, and air is admitted into *E* for the rest of the return stroke.

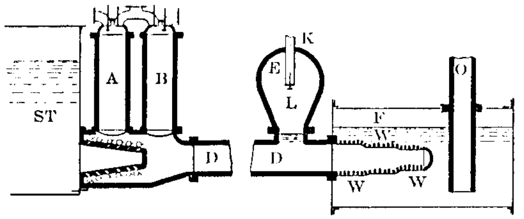


Fig. 44.—Double-barrel Pump with Two Combustion Chambers.

Now it is easy to see that if the pipe *K* is made vertically adjustable with regard to *E*, the point of the cycle at which *L* shuts can be varied, and more or less air entrapped in *E* at will. But the amount of energy stored in this air will also vary with its quantity, for we assume that the degree of compression remains constant, and is indeed fixed by the pressure maintained in *F*. Consequently the ratio of the total energy of the working stroke to the energy stored in the compressed air in *E* can be made anything desired, or, in other words, we can obtain any compression pressure of the new charge in *B* which we like, and this independent of the water lift. The advantage is obvious, for compression pressures equal to those in modern gas-engines can be employed with a corresponding increase in thermal efficiency. Further, by manipulating the position of pipe *K* a given pump can be made to meet any conditions as to height of lift, for if the lift increases *K* can be raised, so that the energy stored in the air in *E* remains the same, there being now less air, but at a higher pressure.

An important development of the arrangement is shown in Fig. 41, and notice is directed to the fact that at each cycle air is drawn into, and rejected from the vessel *E*. Let us suppose *K* to be connected to a supply of combustible mixture instead of opening into the atmosphere, we shall then have an automatic pump for taking in mixture and discharging it under pressure. If the discharge is into a reservoir from which combustion chambers *A* and *B* can be supplied, we have at once a means of quickening the cycles and greatly increasing the output of a given size apparatus. It is convenient to replace vessel *E* by two vessels, one for air and one for gas, so as to maintain the combustible constituents separate until they enter the combustion-chambers. If the first portion of the out-stroke of the water column is allowed to reject the surplus air and gas back to the sources of supply, then the action throughout the cycle is precisely that described when using the single vessel *E*, except that a larger proportion of the total energy is absorbed in the compression of air and gas, but the excess is given out again during the expansion of the pre-compressed charge in either *A* or *B*. The chief advantage arises from the more rapid working, as there is no longer any need to wait for the water-level in *A* or *B* to fall under the action of gravity when the charge is being taken in. In fact, the apparatus becomes practically independent of the water-level on the supply side.

The Humphrey Pump as an Air-compressor.—If the column of water oscillating in the play-pipe of a Humphrey pump is used as a water piston, and caused to rise and fall in an air vessel fitted with suitable valves for the inlet and outlet of air, the combination constitutes an air-compressor of a very efficient type and promising many advantages. Take the case of a single-barrel pump and a single air vessel, shown in Fig. 45. The cycle, which may take two seconds to accomplish, is as follows:

	Pump chamber *A*	Air-compressor chamber *C*
1st out-stroke........	Expansion to atmosphere. Intake of scavenging air.	Expulsion of air till water shuts valve *s*. Compression and discharge of compressed air till water shuts valve g. Cushion till water comes to rest.
1st in-stroke..........	Exhaust till water shuts valve *e*. Cushion till water comes to rest.	Expansion of cushion to atmosphere. Intake of fresh air.
2d out-stroke.........	Expansion of cushion to atmosphere. Intake of combustible charge in excess.	Compression of air, but not sufficient for further delivery.
2d in-stroke..........	Rejection of surplus charge till water shuts valve *r*. Compression of charge till water comes to rest.	Expansion of compressed air.

The flexibility of the air-compressor can now be studied. To begin with the pump side, the level of the inlet valve *e*, and the rejected charge *r*, are assumed to be variable, although Fig. 45 being merely a diagram, does not show how the pipes carrying these valves are moved vertically. As the level of these two valves controls the amount of charge ignited at each cycle, and the amount of the cushion space, their regulation is all that is required to increase or diminish the energy developed per working stroke. On the compressor side the position of the valves *g* and *s* controls the cycle of operations on this side of the apparatus, and renders it possible to compress a large volume of air to a low pressure, or a smaller volume of air to a high pressure, or to make any intermediate changes which may be desired.

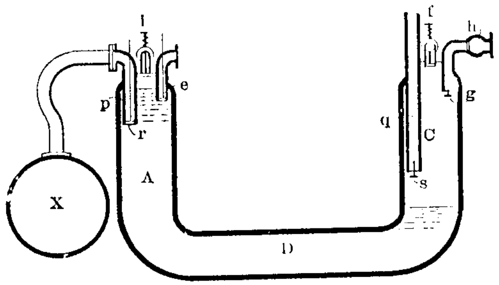


FIG. 45.—Humphry Pump an an Air Compressor.

Thus all the conditions of output, up to the full limit of the compressor, may be governed at will, and for all ranges the compression pressure of the new charge may be kept up to the required degree, so that the apparatus works at its maximum efficiency throughout the whole range. The amount of water which oscillates between the chambers should theoretically be altered along with the total capacity per working cycle, but the reason for this is merely to prevent the last portion of each down stroke from being wasted by taking in surplus combustible mixture in one chamber, or surplus air in the other chamber, to an undue extent. If the surplus of the combustible mixture is unnecessarily large, the extra amount rejected will increase the pressure in the reservoir X, and this increase of pressure may be made to automatically govern a water supply, and so bring up the total volume of reciprocating water, or to allow part of the water already in the apparatus to escape, so as just to keep a small amount of excess charge for each cycle, no matter what may be the output of the pump.

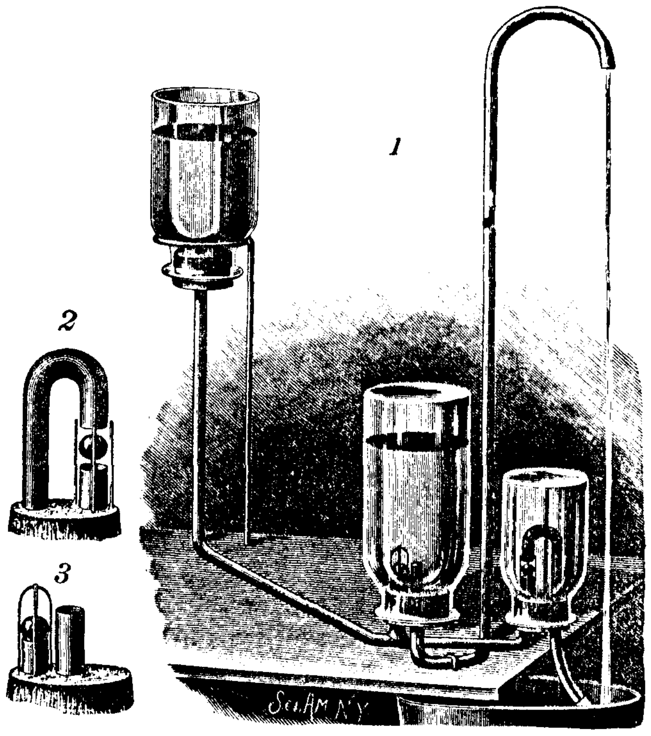
The Humphrey pump is far beyond the experimental stage. It has been built in several units up to 1000 h.p. and its economy and efficiency may be said to be fully established. Four Humphrey pumps, with a capacity of 33,000 gallons per minute each, against a head of 25 feet, have been installed and are in successful operation at Chingford, England, pumping water from the river Lee as a contribution to the water supply of the city of London.

Not much can as yet be said as to any extensive employment of the Humphrey principle to practical air compression, but it seemed at least proper to mention it here as among the practicable possibilites.

turned in opposite directions. All of the tubes are stopped, and the float is placed in a large vessel of water, when the can is filled with water and the stopper of the tube, *a*, is withdrawn, thereby allowing water to escape from the can, and by reaction drive the can backward.

When the straight tube, *a*, remains closed, and the bent tubes, *b*, are opened, the reaction of the issuing streams results in the rotary movement of the apparatus. The apparatus arranged in this way illustrates the principle of Barker's mill.

Fig. 74.



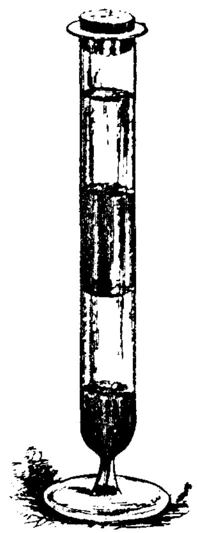
Hydraulic Ram.

The hydraulic ram, a simple form of which is illustrated in Fig. 74, depends for its action on the momentum of the water column and upon the elasticity of air. The reservoir in the present case consists of an inverted glass bottle having no bottom, and provided with a perforated stopper in which is inserted one end of a tube, preferably lead, on account of the facility with which it may be cut and bent. The other end of the tube is branched, one branch extending through a stopper inserted in an inverted bottle which serves as an air chamber. The other branch of the tube extends to the overflow valve. In the stopper of the air chamber is inserted a second tube, which is bent upward and curved over, forming the riser.

The smaller bottle, which serves as a valve chamber, is provided with a stopper which receives the branch of the supply tube and an overflow tube. The arrangement of these tubes is shown in detail at 2, the curved tube being the overflow, the straight one the inlet. To the inlet and overflow tubes is fitted a valve consisting of a metal ball or a marble. The fitting is accomplished by simply driving the ball against the end of each tube, so as to form valve seats. Four wires are inserted in the stopper around the inlet tube to prevent the escape of the valve. The distance which should separate these tubes, as well as the weight of the ball valve, is determined by experiment.

In the air chamber above the branch of the supply tube is confined a ball valve by a cage formed of wires inserted in the stopper, as shown at 3. This valve is fitted in the manner already described.

Fig. 75.



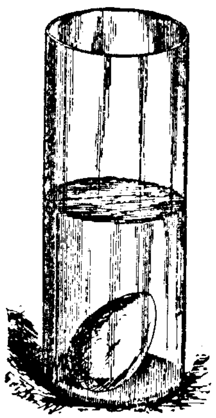
Vial of Four Liquids.

The discharge tube extends above the level of the reservoir. The reservoir and the tubes are supported by wire loops and standards inserted in a base board.

Water flows from the reservoir through the valve chamber and out at the overflow. When the velocity of the flow is sufficient to carry the valve in the valve chamber up against the end of the curved overflow tube, the overflow is immediately checked, and the momentum acquired by the water causes it to continue to flow for an instant into the air chamber, compressing the air in the chamber, and causing the water to rise in the discharge tube. As soon as equilibrium is established, the valve in the air chamber closes and the valve in the valve chamber falls away from its seat on the overflow tube, allowing the water to discharge again, and so on, this intermittent action continuing so long as there is water in the reservoir. The water discharged by the riser is only a fraction of that flowing out of the reservoir.

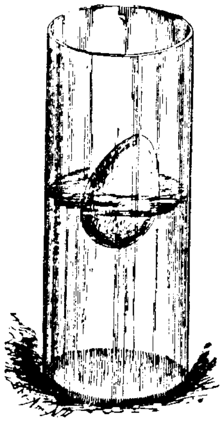
We have already noticed (Fig. 66) that a liquid will assume the same level in communicating vessels. The size and form of the vessels is immaterial. The smaller one may be inclined, curved, or bent in any form and the larger one may have any capacity, still the result will be the same.

Fig. 76.



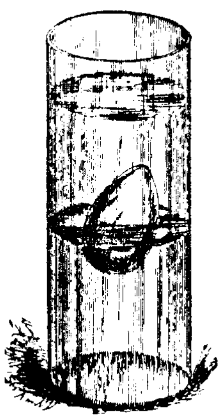
Egg in Fresh Water.

Fig. 77.



Egg Buoyed up by Salt Water.

Fig. 78.



Egg in Equilibrium between two Liquids of Different Densities.

When, however, the vessels contain liquids of different densities, the level will be no longer the same. In such case the lighter liquid will stand higher.

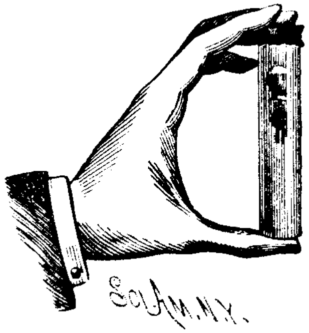
When several liquids of different densities which do not mix are contained in the same vessel, there will be stable equilibrium only when the liquids are arranged in the order of their densities, the heavier liquid being, of course, at the bottom. This is illustrated by the "vial of four liquids," shown in Fig. 75. A test tube with a foot makes a convenient receptacle for the liquids. In the bottom of the tube is

placed mercury. The second liquid in order is a saturated solution of carbonate of potash in water. The third is alcohol, colored with a little aniline red to mark the division of the liquids more clearly. The fourth is kerosene oil. When these liquids are shaken up, they mix mechanically, but when the tube is at rest the liquids quickly arrange themselves in their original order.

The experiment illustrated in Figs. 76, 77, and 78 shows the effects of liquids of different densities. Two pint tumblers or similar vessels are necessary for this experiment. Half fill one with water and the other with strong brine. Into the water drop an egg. It goes to the bottom (Fig. 76). An egg dropped into the brine floats (Fig. 77). By carefully pouring the brine through a long funnel or through a funnel with an attached tube, which will reach to the bottom of the tumbler containing the pure water, the water and the egg will be lifted, and the egg will float in equilibrium at the middle of the tumbler.

The first experiment shows that the egg is a little more dense than pure water, the second that brine is more dense than the egg, and the third that the egg can be supported in equilibrium between two liquids of different densities.

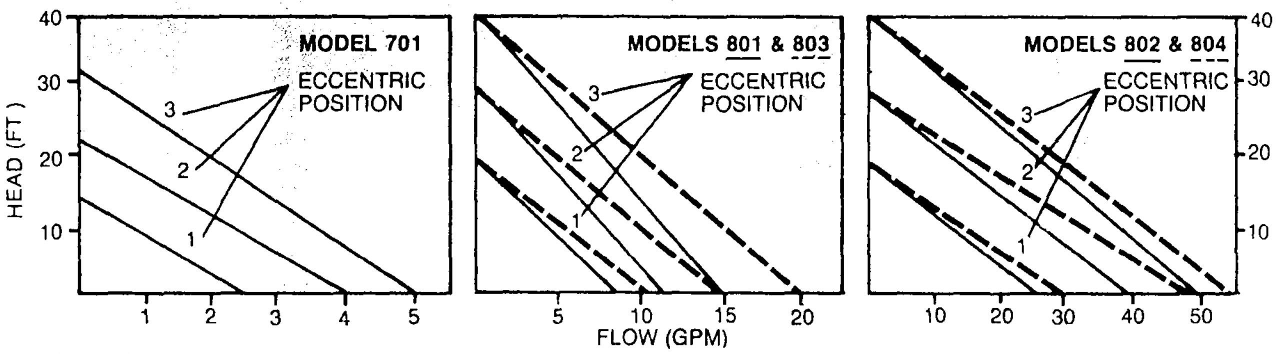
Fig. 79.



The Cartesian Diver.

The hydrostatic toy known as the Cartesian diver illustrates the several conditions of floating, immersion, and suspension in equilibrium. In a tall, slim glass tube, closed at the bottom and filled with water, is placed a porcelain or glass figure having a glass bulb attached to its head. The glass bulb has a small hole in the bottom, and is filled partly with water and partly with air, the proportion of air and water being such as to just allow the bulb to float. The top of the tube is closed by a piece of flexible rubber tied over its mouth. The pressure of the fingers upon the rubber communicates pressure through the water to the air

Three positions on flywheel permit pressure/flow curve selection.



OSCILLATING 'LOOP' PUMPS MOST FLUIDS

Unit can be throttled to zero, run dry

David J. Bak, East Coast Editor

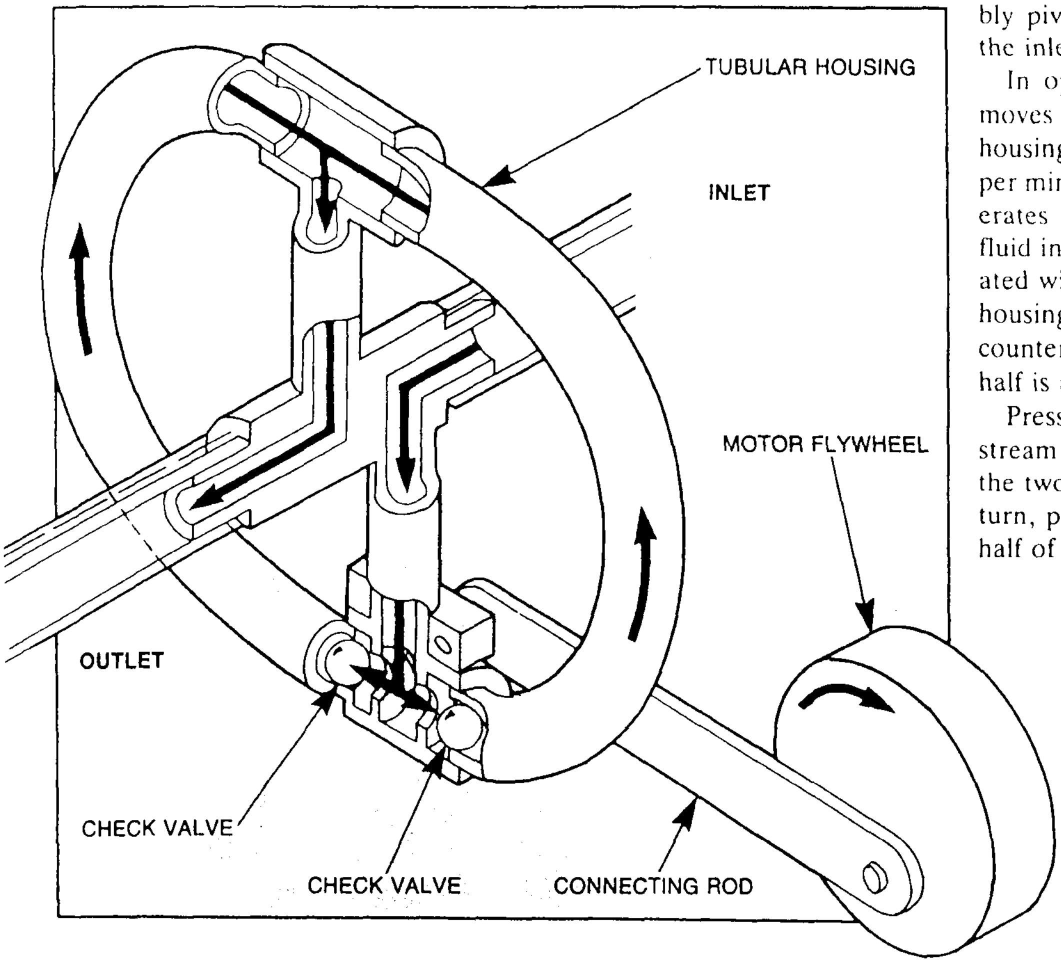
Englewood, CO—The relative motion between this pump's tubular housing and the fluid contained in that housing creates a pumping action free of pistons or impellers. The housing, a fluid-filled loop, oscillates in the general direction of flow. "Inertia" carries fluid to the outlet.

Manufactured by Horst Dynamics Inc., the pump comprises the housing, a drive motor,and connecting rod. The latter, eccentrically positioned on the motor flywheel and fixed to the housing's lower half, generates the oscillating motion.

The housing comprises an axially aligned inlet and outlet, right/left flow paths,and two check valves—one for each flow path. The assembly pivots around the centerline of the inlet/outlet ports.

In operation, the connecting rod moves the bottom portion of the housing back and forth at 1800 cycles per minute. When the housing accelerates clockkwise, the column of fluid in the loop's left half is accelerated with the loop outlet. When the housing changes direction and moves counterclockwise, fluid in the right half is accelerated.

Pressure fluctations just downstream of the valves open and close the two check valves. The valves, in turn, provide free inlet flow to each half of the loop.



Pump components. Unit pivots about centerline of inlet port. Connecting rod, eccentrically positioned on flywheel, transmits oscillating motion. When loop accelerates clockwise, fluid in left half accelerates. When loop decelerates, inertia pumps fluid to outlet. Housing's right half operates in the same manner.

Acceleration forces the fluid column against the downstream side of the check valve, blocking inlet flow. Inertia carries the column from the valve to the outlet during deceleration. Downstream pressure drops below inlet pressure, and the check valve opens.

Because of overlapping pump cycles, delivery is uniform and uninterrupted. Outlet flow may be throttled any amount without damaging the pump—even down to zero. Unlike positive-displacement pumps, no bypass or relief valve is needed. Unlike centrifugal pumps, the fluid will not heat up.

The pump may also be run dry. There are no dynamic seals to burn out; no shafts or bearings cooled by the pumped medium. The only moving parts are the check valves.

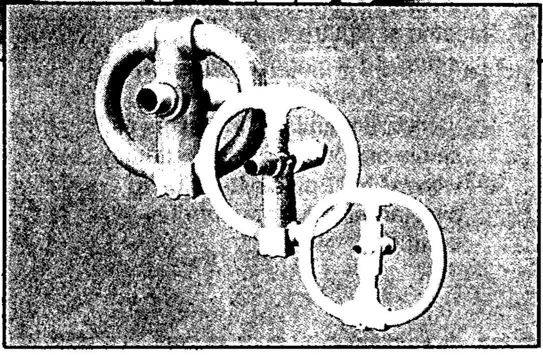
Self-priming and inherently efficient, the pump handles fluids with viscosities up to 2000 cps. The only significant losses are the pressure drop across the check valves,and the energy required to oscillate the housing.

Units are available in CPVC or stainless steel.

Additional details.....Contact Wilbur J. Shaffer, Horst Dynamics Inc.,3960 S. Kalamath, Englewood CO, 80110, 303-761-6309. □

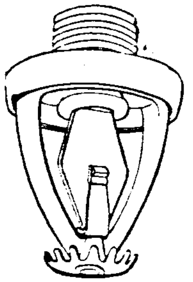
Did you find this article interesting? If so, let us know. Circle No. 846

Self-priming to 8 ft, unit handles most fluid mediums. Motor HP requirements are not affected by viscosity. (Inventor with pump).

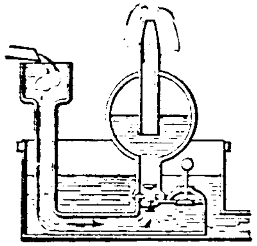


Different size pump elements alter pump capacity. Motor speed (1750 rpm) remains same. ►

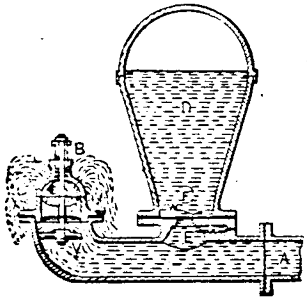
Hiscox, p. 150-151.



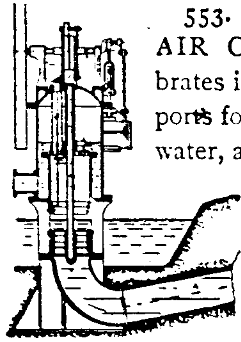
550. AUTOMATIC SPRINKLER.—The valve is held tightly closed by the diamond-shaped post resting on a bell-crank clip, which is held in position by fusible solder, melting at about 200° Fah., at which temperature the solder melts and the pressure casts the clips loose. The star washer scatters the stream.



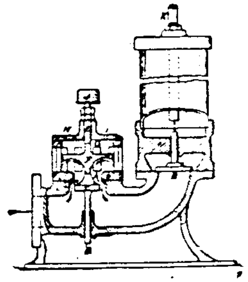
551. HYDRAULIC RAM, the "Montgolfier" idea for a fountain supplied by a water ram.



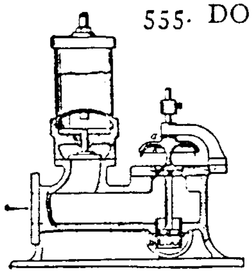
552. HYDRAULIC RAM.—A, Driving pipe; V, impact valve; C, valve-bonnet cage and spindle; W, force valve; F, outlet to force pipe; D, air chamber; E, snifting hole, sometimes furnished with a small ball valve which allows air to draw in at each rebound of the drive-water column, and thus to keep the air chamber supplied with air.



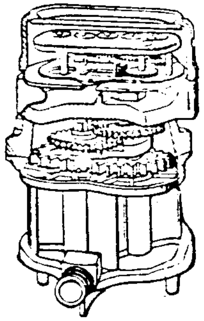
553. "PEARSALL'S" HYDRAULIC RAM AND AIR COMPRESSER.—A hollow or open piston vibrates in a cylinder, perforated all around with escape ports for egress of water. An air chamber receives the water, and the air which is drawn in through the ports, which becomes compressed. A small air motor drives a crank shaft and fly-wheel, which operate the piston. By the sliding motion of the piston in closing the ports, water hammer is avoided, thus enabling the use of a ram of very large dimensions.



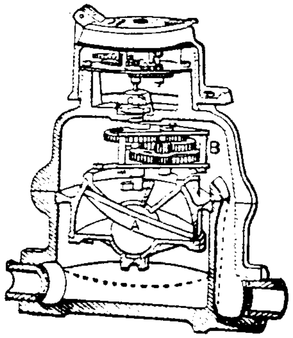
554. SILENT HYDRAULIC RAM.—The curved reaction disc, F, serves to lift the piston valve, C, quickly without shock. The air cushion at G stops the lift at the moment of closure of piston valve, C. J, a stop set-screw; H, valve cage; B, force valve; K, force pipe; I, vent hole to air cushion.



555. DOUBLE-PISTON REACTION HYDRAULIC RAM.—The two pistons, B and O, are on the same spindle with curved reaction disc, A. G is a leather washer to soften the contact with guide yoke. The cage at D guides the lower piston and serves to increase greatly the freedom of water-flow from the drive pipe, thereby increasing the duty of the ram.

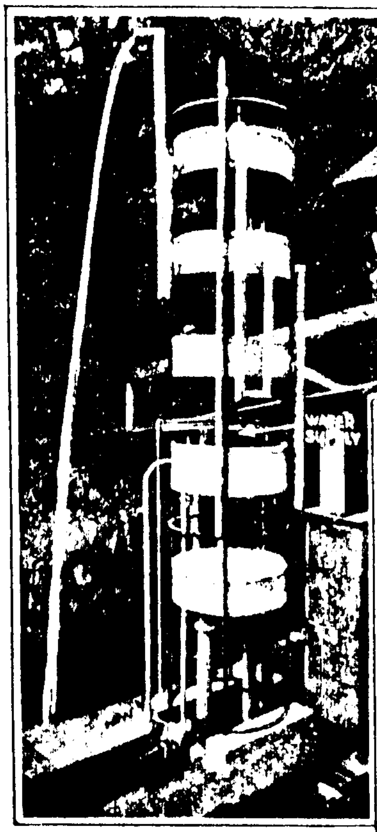


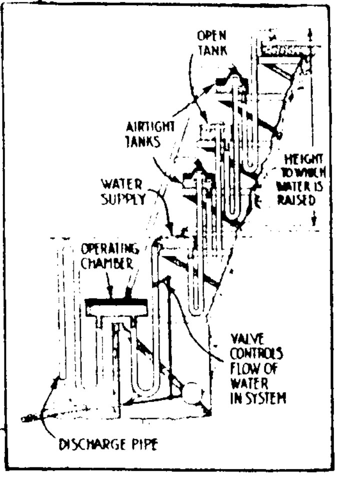
556. WATER METER.—"Union" water meter model. The water passes through a rotary motor with equalizing gear, from which the dial pointers are driven by a clock train and counter.



557. DISC WATER METER, "Hersey" model. The disc piston, A, oscillates by the passage of water through the disc chamber. The spindle of the disc, by its oscillating movement, rotates the crank and gears of the index-wheel train.

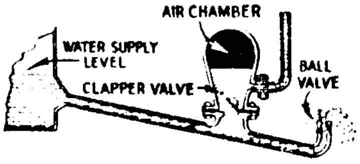
SERIOUS problems of irrigation and rural water supply may shortly be solved by an extraordinary apparatus recently perfected in England, which automatically raises 14 gallons of water from a stream to a height of 20 feet in a little less than three minutes, without the use of pumps or any source of power other than what nature furnishes. This device, seeming at first glance something like perpetual motion applied to water, and called by its inventor, T. G. Allen, a "hydrautomat," or "water staircase," is hailed by engineers as marvelously ingenious. It is an extraordinary improvement on the hydraulic ram commonly used.




OPEN TANK
AIRTIGHT TANKS
HEIGHT TO WHICH WATER IS RAISED
WATER SUPPLY
OPERATING CHAMBER
VALVE CONTROLS FLOW OF WATER IN SYSTEM
DISCHARGE PIPE

Water "Lifts Itself"

Arrangement of alternating open and closed tanks, one above the other, in the experimental "water staircase," installed near London, England, is shown above, while the diagram at the right explains how water, with the aid of air pressure, is made to "lift itself by its bootstraps" to twice the height of its original level



In the hydraulic ram (above), some of the water is forced through a clapper valve into an air chamber, compressing the air. This closes the valve and forces the water to a higher level

Two Energy Sources Used

In all the centuries in which men have attempted to make nature serve them through two sources of energy—the weight of a column of water and atmospheric pressure—no such radical invention has been recorded.

Using these two forces, the experimental "water staircase"—erected near London—is said to operate with extraordinary efficiency. It consists of an alternating series of open and closed tanks, one above the other, the action of which is to use the energy of a quantity of water at a given level to raise a smaller quantity of water to a higher level.

The stream from which the power is derived, empties into an open intake tank located 7½ feet above the lowest level of the device. Thus there is an available fall of 7½ feet from the upper level of the stream, which may be termed the head race, to the lower level, or tail race. Halfway between the upper and lower levels is an airtight operating chamber, supplied with an intake pipe from the head race, and a discharge pipe to the tail race. Above the upper water level is the alternating series of closed and open tanks. These tanks are interconnected by pipes. In addition, the closed tanks are coupled to the operating chamber by an air pipe.

Operation of the "water stairway" is confined to two strokes—a pressure stroke followed by a suction stroke. The pressure is created by the water column flowing from the open supply basin just below the level of the head race into the airtight operating chamber.

The effect of this water flow is to compress the air in the operating chamber and to force it upward through the air pipe into the two closed and water filled tanks. Immediately the water in these tanks, lifted there by the preceding stroke, is forced upward into the two open tanks above them.

Thus at the end of the pressure stroke, the operating chamber and the two open tanks are full of water, while the two closed tanks are full of air.

How the Water "Climbs"

In the suction stroke the contents of the operating chamber are discharged downward into the tail race, or discharge pipe, through a siphon, and at the same time the inlet from the head race into the operating chamber is automatically closed. This is accomplished by a valve actuated by the rush of water out of the discharge pipe. In the operating chamber is thus produced a vacuum that also extends to the two closed tanks, by virtue of the connecting pipe.

The result of this vacuum is that it sucks up the water "one flight" from the corresponding open tank below. Thus, at the end of the suction stroke, the open tanks are empty of water and the closed tanks are full. The valve in the head race inlet then opens automatically, pressure water is admitted into the operating chamber, a new pressure stroke starts, and the cycle is repeated.

Compressed Air Drives Hand Shovels into Hard Ground

Above: How pneumatic shovels are used in underground excavation, where working space is limited. Note compressed-air pipes attached to shovel handles

At right: A typical air-driven shovel as it is used in trench digging

HAND shovels driven by air are greatly facilitating the digging of tunnels and other types of excavations with less fatigue to the laborers. These shovels are particularly successful in cutting up hard and soft clays, or wherever the ground is not hard enough to warrant drilling and blasting.

Spade Operated by Air-Driven Piston

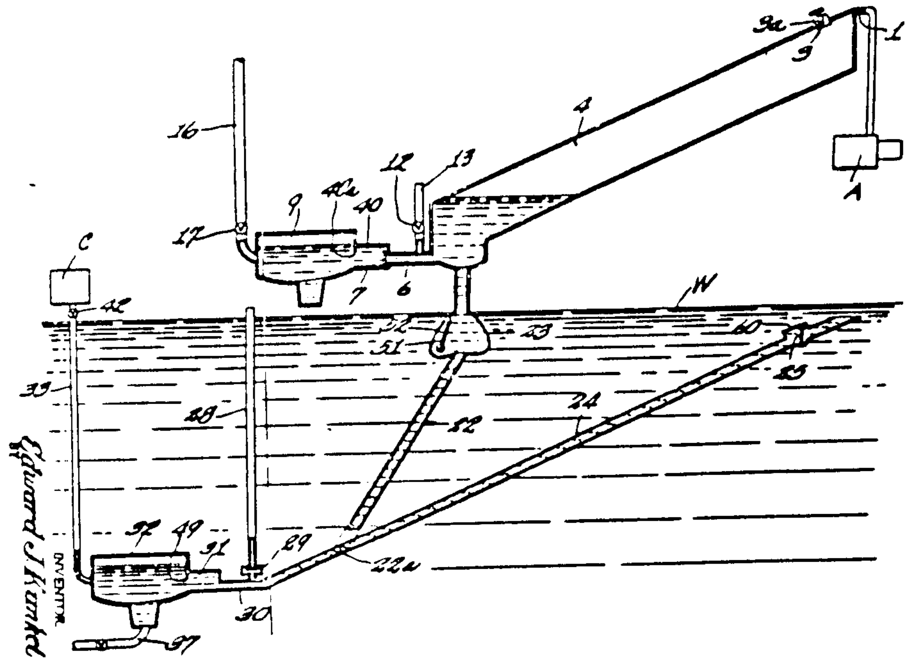
The operation is practically the same for all types of these shovels. The spade is inserted into a cylinder forming the lower end of the shovel handle. In this cylinder a piston driven up and down by the jet of compressed air strikes the end of the spade shank and gives it an impulse that drives the blade into the hardest earth.

To prevent shock to the hands of the laborer in case the shovel is not in contact with the ground when air is admitted to the cylinder, a spring on the top of the spade shank absorbs the blow.

These shovels have shown surprising adaptability to underground work where space is at a minimum or where the necessity for speed demands the labor of several workmen in small spaces.

RESEARCH FILE

Pharoahs Pump



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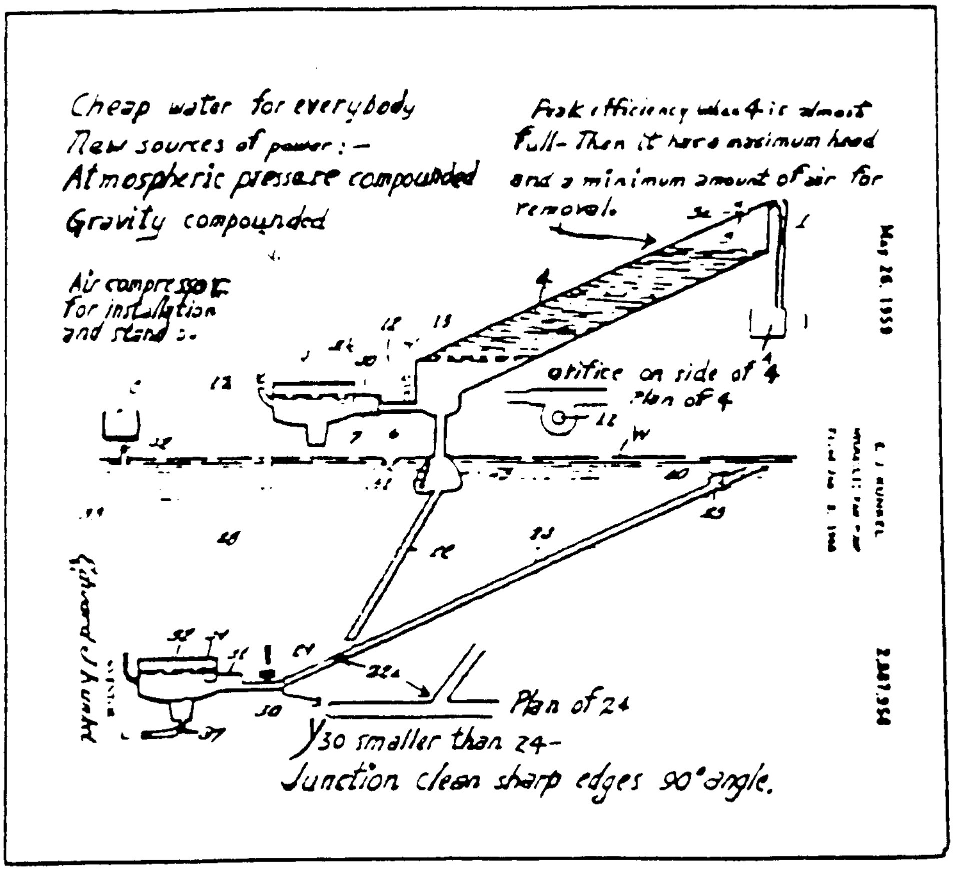
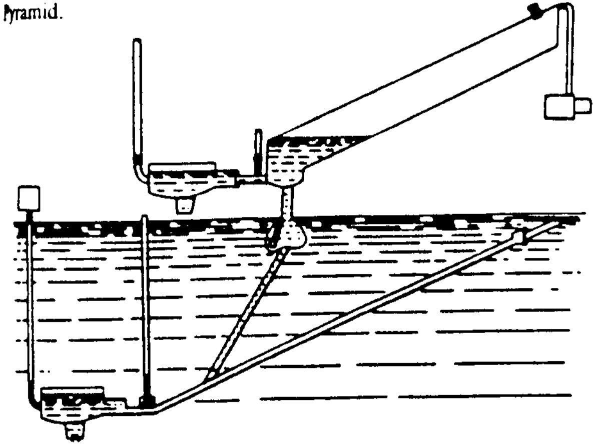

Cheap water for everybody
New sources of power: –
Atmospheric pressure compounded
Gravity compounded
Air compressor for installation and stand-by
Peak efficiency when 4 is almost full – Then it has a maximum head and a minimum amount of air for removal.
orifice on side of 4
Plan of 4
Plan of 24
30 smaller than 24 –
Junction clean sharp edges 90° angle.
May 26, 1959
E. J. KUNKEL
2,887,956

Edward J. Kunkel, 295 W. Market St., Warren, Ohio 44481, says he has investigated almost every inch of the interior of the Great Pyramid. He suspected it was actually a giant ramp pump and that the pyramid was built by floating the blocks on barges on the rising water level in the interior of the Pyramid, as the name pyramid implies. In Greek, "pyr" means water, and "amid" means within. On page 30 of his self-published book, "The Pharaohs's Pump", Mr. Kunkel included a diagram of what is known as the well. He feels certain this was the air combustion chamber for the ram pump, which system would have included all the other passages as well. According to Mr. Kunkel the shape of the well had the following effect: Air trapped near the small ceiling formed a cushion, and the moment a downward flow occurred, air pressure in the cell conversely became extremely high and, seeking relief, moved the water between the cell and the chamber, thus initiating the action which a moment later developed into a whirlpool. A model worked better with this cell than without it. Was it knowledge or accident? Inspired by what he found in the Great Pyramid, Kunkel invented a ram pump and obtained a U.S. Patent on it.

Pharoahs Pump. Back in the early 1930's Edward Kunkel became interested in how the great pyramids were built. In the course of his research over the years, astonishing indications and facts were uncovered. The evidence all pointed to the fact that the great original pyramid was a giant water pump, and all the chambers were exacting parts for its functioning. Working models of this pump design proved the Kunkels contentions from the '40's to the '60's. Despite a U.S. Patent and overwhelming logical and physical evidence, the inventor was ignored. Dozens of government agencies, research centers, and universities were contacted and failed to show the slightest interest. Finally Kunkel printed a little book on the whole subject. It told how the pyramid was really built, how a phenominal amount of water was pumped at practically no cost, and how each portion of this amazing pump functioned. When, with even the slightest effort, such a pump design could furnish free water to the parched lands and peoples of the world, the author could not explain its rejection.

→ Book:
$17.50 + $2.50 postage & handling.
David French, POB 2010 - CET,
Sparks NV 89432 USA

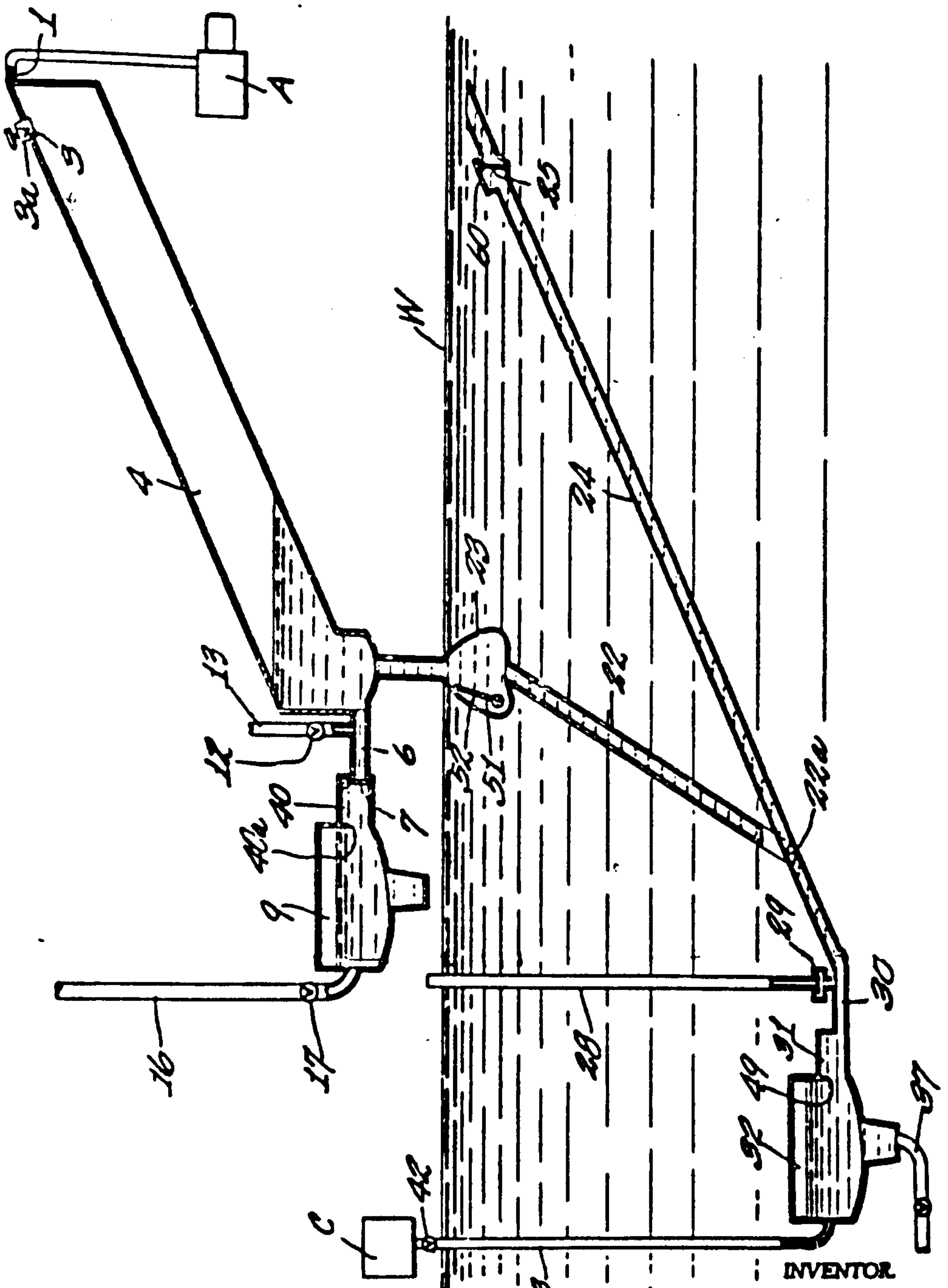
Sketch of Edward J. Kunkle's "Pharoah's Pump," U.S. Patent 2887956; a practical theory for the hydraulic engineering process possibly used in building the Great Pyramid.



May 26, 1959 E. J. KUNKEL 2,887,956

HYDRAULIC RAM PUMP

Filed Jan. 3, 1955



INVENTOR

Edward J. Kunkel

BY

United States Patent Office

2,887,956
Patented May 26, 1959

2,887,956

HYDRAULIC RAM PUMP

Edward J. Kunkel, Warren, Ohio

Application January 3, 1955, Serial No. 479,428

4 Claims. (Cl. 103—77)

This invention relates to a pump, more particularly a ram pump. My invention provides economy in pump operations and movement of liquid containing fine solid matter without undue damage to the working parts of the pump, and the principal object of this invention is to provide new and improved pumps of this character.

Whereas the common water ram is actuated by the starting or stopping of a waste flow which is diverted into a chamber of compressed air and locked in by a check valve, the compressed air forcing the water in the ram to a higher level, my improved ram pump is not actuated by the wasting of water, but rather by the changing of air pressure on the surface of an upright water column, which is done by producing a vacuum on the top of the column, the changing pressure causing an up and down movement of the water column.

The downward movement of the water in the tube containing the water column is limited by a buoyant flap valve disposed within the tube. When the water is at rest the flap valve remains open, but a downward movement through the tube closes the flap valve, thus trapping water above the valve.

The water tube is connected to a submerged diagonal tube, and the column of water in the diagonal tube is supported by a cushion of air. An upward movement of water through the water tube causes the column of water in the diagonal tube to be set in motion. The compressed air cushion absorbs the energy developed by the downwardly moving diagonal column of water and in turn forces the water in the water tube to a higher level so as to discharge such water into a diagonal tube above the ground. Whereas the head in the common water ram is constant, the head in the diagonal tube above the surface must be constantly built up, since it discharges itself during each cycle. If the head in the diagonal tube above the surface is too low to discharge, another cycle of operation will build it up to a height that it will discharge to a predetermined level. It should be noted that the length of the lower or submerged diagonal tube is equal to or or greater than the vertical height of the pump.

The design and operation of my improved ram pump embodies the economy of operation of the common water ram which uses the force of gravity to do work and my improved pump may be used in situations where it is impossible to use the ordinary ram pump. For example, the ordinary ram pump will not work at a level where the waste water cannot drain away and its uses are thus limited.

My improved ram pump may be installed in a stream or reservoir and take water from it, and should be quite useful in supplying irrigation canals, providing cheap water for the culture of rice, and removing excess water in regions where there is a superabundance, such as in the Netherlands. My improved pump may also be used in regions where the water is extremely turbid and is laden with sludge and abrasive silt, without the necessity of constant maintenance and replacement of its moving parts.

Physical facts and calculations point out that in a pump of my improved type, the efficiency increases in a geometric ratio with the pump size, a limiting factor being that as the size increases the solubility of air in water increases also, due to the extremely high air pressures that come in contact with the turbulent water.

As an example, a diagonal tube an inch square and 60 inches long with a pitch of one foot fall or every two linear feet exerts a static pressure of one pound on the bottom of the tube; a ratio of one pound to 60 cubic inches of mass. But, if every dimension is multiplied by 2, the ratio is 2 pounds to 480 cubic inches of mass; if the dimensions are multiplied by 4, the ratio is 4 pounds to 3840 cubic inches of mass; and so on.

In the drawing accompanying this specification and forming a part of this application, there is shown, for purposes of illustration, an embodiment which my invention may, and in this drawing the single figure discloses my improved ram pump in schematic form.

The improved ram pump herein disclosed comprises two tubes 24 and 4, the tube 24 being the lower one and being submerged below the level of a body of water W, and the tube 4 being the upper one and being disposed above the level of the body of water. Each of the tubes 24 and 4 is disposed at an inclination to the horizontal and may be termed a diagonally disposed tube. Each tube is preferably rectangular in cross-section and the tubes are disposed one above the other and their respective lower ends are connected by an upright conduit 22.

The tube 4 has an opening 1 at its upper end which is adapted to be connected to means A for establishing a vacuum in the tube 4. Such means may take the form of a standard motor-driven vacuum pump and therefore need not be described in detail. The tube 4 is also provided with an opening 3, controlled by a valve 3a, for selectively admitting air under atmospheric pressure to the tube.

The tube 24 is substantially uniform in cross-section except at a lower horizontal portion 30, which latter portion is reduced in cross-section. Near its upper end, the tube 24 is formed with an enlargement 60 in which is disposed a hinged check valve 25. The enlargement is of sufficient size to permit the valve to swing upwardly to its completely open position and to thus permit water to freely enter the upper end of the tube 24 and flow downwardly therein. The check valve 25 is moved to closed position, against a force normally biasing the valve to open position, by water tending to flow in a direction upwardly of the conduit 24 and therefore acts to automatically prevent upward discharge of the water from the tube 24.

Two large air compression chambers 9 and 32 are disposed adjacent to and are in communication with respective lower ends of the tubes 4 and 24. The chambers may be formed by any suitable means and therefore their specific construction need not be described.

An auxiliary air compression chamber 40 communicates with one end of the chamber 9 through an opening 40a and the other end of the auxiliary chamber 40 communicates with the lower end of the tube 4 by means of a small conduit 6. A conduit 13 establishes communication between conduit 6 and the atmosphere whereby any reduced pressure within the tube 4 or the compression chamber 9 may be relieved at will by proper manipulation of a valve 12 interposed within the conduit 13.

A water discharge pipe 16 is in communication with the chamber 9 for discharging water therefrom, the discharge from pipe 16 being controlled by a valve 17. A check valve 7 in the auxiliary compression chamber 40 permits flow of water from tube 4, through pipe section 6 and to air compression chamber 9, but prevents a reversal of water flow.

The lower horizontal portion 38 of lower tube 24 communicates with one end of an auxiliary air compression chamber 31, the other end of the latter communicating with the chamber 32 through an opening 49. An upright pipe 33 communicates with the chamber 32 and is connected with a source of compressed air C, a valve 42 controlling flow through the pipe 33. The valve 42 may be opened to supply air under pressure to the chamber 32 when necessary.

An upright pipe 28 is in communication with the lower horizontal portion 30 of the lower tube 24 and a disk check valve 29 in the tube permits a low pressure condition within the portion 30 to unseat the valve 29 and provide for flow of air under atmospheric pressure to the portion 30. The chamber 32 may have a valve-controlled clean-out conduit 37 at its lower end to provide for selective flushing of the chamber of dirt and silt.

The lower tube 24 has a side opening 22*a* for communication with the lower end of the conduit 22. An enlargement 23 is formed in the upper portion of the conduit 22 and a hinged, buoyant valve 51 is disposed within the enlargement. The valve is in the open position shown when the water in the pump is at rest, and is automatically moved to closed horizontal position when water flows downward through the enlargement 23. An aperture 52 in the valve 51 provides a bleeder hole to prevent the buoyant valve from being locked in closed position. The aperture also reduces water hammer in the conduit 22.

In operation, and assuming water has entered the tube 24 and stands in conduit 22 substantially level with the level of the water body W, when pressure is reduced in tube 4, as by operation of the vacuum pump A, water from conduit 22 will rise in tube 4, thus providing for further water flow into tube 24 and conduit 22.

The energy of the water flowing downwardly in tube 24 is absorbed by compression of the air in chambers 31 and 32 and, since the valve 25 prevents upward flow of water in tube 24, the air pressure in chambers 31 and 32 therefore acts on the water in conduit 22 to urge the water upwardly into tube 4.

The operation of vacuum pump A may be in timed relation with operation of other portions of the pump so that air pressure in tube 4 is reduced to draw water into the tube to a predetermined level, whereupon the valve 3*a* is opened to admit air under atmospheric pressure to the tube. Such action causes the level of the water in tube 4 to drop but as the column of water tends to move downwardly, the valve 51 is thereby closed so that the lowering level of water in tube 4 flows into chambers 9 and 40 and compresses air in the upper portions thereof.

When valve 17 is opened, the compressed air in chambers 9 and 40 will expell the water from the chambers so that it may be directed to good use by the conduit 16.

As water is drawn upwardly into tube 4 by reduced air pressure therein, it will be appreciated that water will flow from the body W downwardly into tube 24 so that there is a constant rise and fall of the water column in conduit 22, with the fall being interrupted by closing of the valve 51 to divert the falling water into the chambers 9 and 40.

In view of the foregoing it will be apparent to those skilled in the art that I have accomplished at least the principal object of my invention and it will also be apparent to those skilled in the art that the embodiment herein described may be variously changed and modified, without departing from the spirit of the invention, and that the invention is capable of uses and has advantages not herein specifically described, hence it will be appreciated that the herein disclosed embodiment is illustrative only, and that my invention is not limited thereto.

What is claimed is:

1. A ram pump, comprising two elongated tubes, each positioned at an angle to the horizontal and said tubes being disposed one above the other, means for establishing a vacuum in the upper of said tubes, a conduit connecting respective lower ends of said tubes, the upper end of said lower tube being in communication with a head of water, a first check valve in the lower tube permitting flow of water from said head downwardly in said lower tube, said check valve automatically closing to prevent return flow of water, a first air compression chamber means communicating with the lower end of said lower tube, flow of water downwardly in said lower tube acting as a piston to increase air pressure in said chamber means to an amount to reverse the flow of water and thereby close said first check valve, thus water thus trapped in said lower tube being elevated through said conduit by the air pressure in said chamber means to enter the lower end of said upper tube, a second check valve in said conduit and permitting water flow through said conduit and into said upper tube and preventing return flow of water through said conduit when the pressure of air in said chamber means is insufficient to force water through said conduit, a second air compression chamber means in communication with the lower end of said upper tube, air being compressed in said second air compression chamber means by the head of water trapped within said upper tube, a liquid discharge outlet in communication with said second air compression chamber means, and means intermediate and in communication with said second air compression chamber means and the lower end of said upper tube for controlling discharge of water.

2. The structure of claim 1 wherein said lower tube has an enlargement adjacent to its upper end, a hinged check valve within said enlargement and hanging vertically when in closed position, an orifice in the side wall of said lower tube adjacent its lower end to establish communication with said conduit, and said lower tube having a reduction in cross-section adjacent to its lower end.

3. The structure of claim 2 and further including an auxiliary air compression chamber means in communication with said first air compression chamber means and with the lower end of said lower tube and positioned therebetween.

4. The structure of claim 3 and further including a short horizontal tube establishing communication between the lower end of said lower tube and said auxiliary air compression chamber means, an air tube communicating with and for admitting air into said short tube, and a third check valve for controlling flow of air through said air tube.

References Cited in the file of this patent

UNITED STATES PATENTS

1,364,617 Daynix ---------------- Jan. 4, 1921

FOREIGN PATENTS

348,365 Great Britain ----------- May 14, 1931

An Analysis of the Hydraulic Ram

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Late Professor of
Mechanical Engineering,
Univ of Calif.,
Berkeley, Calif.
Mem. ASME

An hydraulic ram is a pump in which the momentum of a driving stream of water undergoing a small head drop is used to pump a small portion of the stream to a head considerably greater than that of the supply. An attempt has been made to identify the features of the ram; the drive head and flow, the discharge (or pump) head and flow, the cycle frequency, and the system efficiency. Analytical results are expressed in terms of the ratio of the maximum drive velocity at drive valve closure to the maximum steady state velocity with the drive valve wide open. This ratio is always less than one. Comparison of the analysis with experimental results gives a numerical correlation of heads and flows. The cycle frequency and system efficiency are not predicted with great accuracy, but the trends are predicted correctly.

Introduction

The hydraulic ram is a rather unique pump, consisting only of pipes and two disk valves. The ram operates with a drive flow from a low head to deliver a pumped flow to a much higher head than the drive flow head. The flows are intermittent and once started, the ram operates continuously with no other external input.

One usual application of the ram is in a stream with a low head dam to utilize the head drop of a few feet for the drive flow. The pumped flow is then delivered to many feet above the stream for water supply to a tank for domestic and irrigation purposes. No electrical or other energy supply is needed; the low head is the drive energy.

There have been many attempts to predict the performance of the ram; a few are listed in the references [1–4].[1] But, for the most part, the analyses are very complex, including water-hammer effects, head loss details, and related effects.

This now-proposed analysis does include the effects of water-hammer and head losses, but in such a simplified form that all effects result in simple dimensionless ratios from which the ram performance is completely identified.

The Hydraulic Ram Action

The hydraulic ram is a pump in which the momentum produced by a flow from a low head is used to pump fluid to a head higher than that of the supply. There are only two moving parts of the pump, a drive valve and a pump valve, Fig. 1, both of which operate automatically from the fluid dynamic actions of the pumping cycle.

The pumping action can be understood by considering the following sequence of events:

(*a*) Both drive valve and discharge valve are closed, the drive valve is held closed by the pressure from the supply head, *H*, and the discharge valve is held closed by the pressure at the pump discharge.

(*b*) The pumping action is initiated by forcing the drive valve away from its seat. The pressure at (2) then becomes essentially atmospheric. Under the action of the head, *H*, the fluid in the drive line is accelerated, flowing out around the drive valve to a stream at a lower level.

(*c*) With proper design of the drive valve, the drive valve will close automatically at some flow velocity past the valve which will produce a pressure distribution sufficient to overcome the weight of the valve. The closure will be very rapid.

(*d*) The sudden closure of the drive valve initiates a pressure at the drive valve that is larger than the static supply head pressure. This pressure opens the discharge valve.

(*e*) The inertia of the flowing mass of fluid in the drive line maintains the flow against the pump discharge head (or pressure in the discharge surge tank) but with decreasing velocity due to the retarding force of the discharge head which is greater than the supply head. During this pumping interval with deceleration of the flow in the drive line, the pressure in the region of the drive valve is approximately the discharge pressure, thus keeping the drive valve closed.

(*f*) When the flow velocity through the discharge valve becomes zero, the discharge pressure reverses the flow through the discharge valve and also in the drive line. The flow reversal closes the discharge valve.

(*g*) Since the flow in the drive line is now moving toward the supply reservoir, and the discharge valve is closed, the pressure in the region of the drive valve decreases rapidly until the combined effect of the atmospheric pressure acting on the drive valve and the weight of the drive valve and the internal pressure acting on the drive valve produces a net force to open the drive valve automatically. (Note here that the drive valve was forced

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[1] Numbers in brackets designate References at end of paper.

Contributed by the Fluids Engineering Division for publication in the JOURNAL OF FLUIDS ENGINEERING. Manuscript received at ASME Headquarters, January 3, 1975. Paper No. 75-FE-F.

open in part (*b*) of this description to initiate the sequence of events. However, once the series of events has taken place, the drive valve opens automatically without any external stimulus.)

(*h*) When the drive valve opens, the pressure in the drive line in the region of the drive valve becomes atmospheric. The supply reservoir head then acts to decelerate the back flow to zero velocity, and then to accelerate the flow for the start of another cycle as in (*b*).

With the usual rapid action of opening and closure of the drive valve and the discharge valve, pressure pulsations are superimposed on the major effects of steady pressure differences and fluid column inertia. These pressure fluctuations produce compression waves and superimposed velocity changes with propagations at the sonic velocity. When all effects are considered, the detailed analysis becomes quite complex.

However, an approximate analysis may be made assuming that the pressure and velocity fluctuations tend to produce average effects caused by the supply reservoir head, atmospheric pressure in the drive line at the drive valve when the drive valve is open, and the discharge head in the drive line when the discharge valve is open.

Pressure and Velocity-Time History

High-sensitivity recordings of pressure-time histories of the pressure in the drive line at the location of the drive valve support the approximations of averaging the fluctuations, Fig. 2. The velocity-time history can be deduced from the pressure-time history and the force-inertia interactions, Fig. 3. The letters identifying features of the velocity refer to events as described in paragraphs (*a*) to (*h*).

Pump Performance

Since the drive line is a necessary feature of the pumping action it must be considered part of the pump. The pump then is the system from (0) to (4) in Fig. 1.

Although both the drive flow and the pumped flow are intermittent, average flow rates can be defined from the total volume of the drive fluid and the total volume of the pumped fluid over the time of a large number of cycles.

The drive fluid uses the head, H, from (0) to (3). Thus the mechanical energy used to drive the pump is

$$P_T = Q_T w H \tag{1}$$

where Q_T is the average flow rate through the drive valve.

The pumped fluid is originally at the surface elevation of the supply reservoir, z_0. The discharge head is the combination of pressure head and elevation in the discharge surge tank, or

$$h_4 = \frac{p_4}{w} + z_4 \tag{2}$$

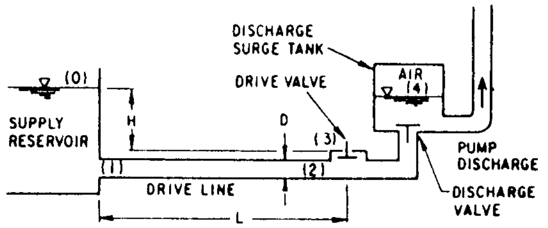

Fig. 1 The hydraulic ram

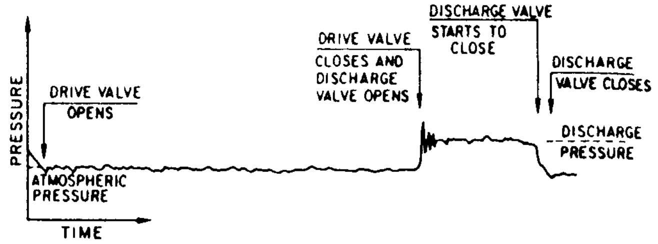

Fig. 2 Typical measured pressure-time history before the drive valve

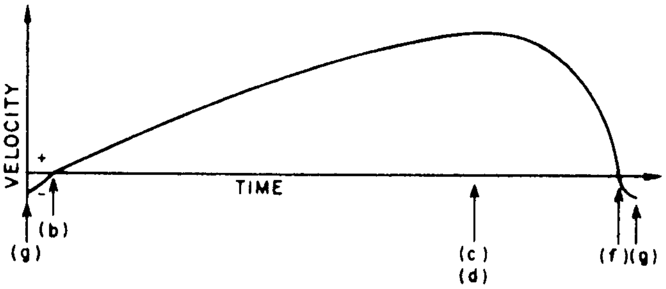

Fig. 3 Deduced velocity-time history in the drive line

The pump head is then

$$h = \frac{p_4}{w} + z_4 - z_0 \tag{3}$$

in which p_4 is the gage pressure since the pressure at (0), which is atmospheric, is taken as zero.

The mechanical energy rate added to the pumped fluid is

$$P_P = Q_P w h \tag{4}$$

where Q_P is the average flow rate through the discharge valve into the surge tank and then into the discharge system.

The efficiency follows,

$$e = \frac{P_P}{P_T} = \frac{Q_P w h}{Q_T w H} = \frac{Q_P h}{Q_T H} \tag{5}$$

Nomenclature

A = cross-section area of flow
D = diameter
e = efficiency
f = friction factor
g = gravitation attraction per unit mass
h = pump head
H = drive head
K = head loss coefficient
L = pipe length
M = combined drive flow head loss factor
N = combined pump flow head loss factor
p = pressure
P = power
Q = volume rate of flow
t = time
V = average velocity
w = weight per unit volume
z = elevation

Subscripts

0, 1, 2, 3, 4: flow positions (Fig. 1)
d = drive valve loss coefficient
e = entrance loss coefficient
m = the drive line velocity when the drive valve closes
p = pump valve loss coefficient
P = pump identification
s = steady-state drive valve velocity with the drive valve wide open
T = drive identification

An interesting analogy can be made. The drive fluid could be used in a turbine to obtain a shaft power output utilizing the flow rate Q_T and the head, H. With the turbine output driving a mechanical pump furnishing the flow rate Q_P at the head, h, a similar effect would be obtained as is obtained with the hydraulic ram. The efficiency of the turbine-pump combination would be given by equation (5). A direct comparison of the efficiencies of the two different systems would be of interest.

Analysis

With the assumption that the effects of the short period pressure fluctuation can be ignored, the hydraulic ram can be analyzed considering only the time dependent effects caused by the average drive head and the average discharge head.

Also, since it is judged that almost all of the period of a complete cycle of events is composed of the period of the acceleration of the drive flow from zero velocity to the velocity at which closure of the drive valve takes place and the period of deceleration of the pumped flow from the time of simultaneous opening of the discharge valve with closure of the drive valve until the pumped flow velocity reaches zero with closure of the discharge valve, this period will be taken as the complete cyclic time interval. In effect, the time from (f) and (g) to (h) will be ignored. Thus the cycle is reduced to two intervals, the drive flow acceleration interval from (b) to (c), and the pumped flow interval from (c), or (d), to (f). These time-dependent flows which may be treated by means of the usual one-dimensional unsteady flow approximation,

$$-\frac{dp}{w} - dz - \frac{fV^2}{2gD}\,dl = \frac{dV}{dt}\,\frac{dl}{g} \tag{6}$$

With constant friction factor and constant diameter of the drive line between (1) and (2)

$$\frac{P_1 - P_2}{w} + z_1 - z_2 - \frac{fLV^2}{D2g} = \frac{L}{g}\frac{dV}{dt} \tag{7}$$

Between (0) and (1) the mass of the fluid being accelerated can be taken to be small compared to the mass of the fluid in the drive line. The steady flow equation then applies with an entrance loss coefficient K_e

$$\frac{P_0 - P_1}{w} + z_0 - z_1 + \frac{V_0^2}{2g} - \frac{V_1^2}{2g} - K_e\frac{V_1^2}{2g} = 0 \tag{8}$$

For the drive flow interval t_b to t_c, the drive valve is assumed to be fully opened, with a loss coefficient K_d. Also, the mass of fluid accelerating between (2) and (3) is small compared to the mass of fluid in the drive line. The steady flow equation applies,

$$\frac{P_2 - P_3}{w} + z_2 - z_3 + \frac{V_2^2}{2g} - \frac{V_3^2}{2g} - K_d\frac{V_3^2}{2g} = 0 \tag{9}$$

Adding equations (7), (8), and (9), noting that $P_0 = P_3 = 0$, and $V_0 = 0$, and $V_1 = V_2 = V$ gives

$$z_0 - z_3 - \frac{V_3^2}{2g} - \frac{V^2}{2g}\left(\frac{fL}{D} + K_e + K_d\right) = \frac{L}{g}\frac{dV}{dt} \tag{10}$$

Furthermore, V_3 can be expressed in terms of the general V in the drive line, $A_3V_3 = AV$ where A is the area of the drive line. Also, $z_0 - z_3 = H$. Then the equation for the drive flow interval becomes

$$H - \left(\frac{A^2}{A_3^2} + f\frac{L}{D} + K_e + K_d\right)\frac{V^2}{2g} = \frac{L}{g}\frac{dV}{dt} \tag{11}$$

Defining, $M = \left(\frac{A}{A_3}\right)^2 + f\frac{L}{D} + K_e + K_d$, then equation (11) becomes

$$H - M\frac{V^2}{2g} = \frac{L}{g}\frac{dV}{dt} \tag{12}$$

Similarly, for the discharge interval (c) to (f), between (1) and (2)

$$\frac{P_1 - P_2}{w} + z_1 - z_2 - f\frac{L}{D}\frac{V^2}{2g} = \frac{L}{g}\frac{dV}{dt} \tag{13}$$

Between (0) and (1)

$$\frac{P_0 - P_1}{w} + z_0 - z_1 + \frac{V_0^2}{2g} - \frac{V_1^2}{2g} - K_e\frac{V_1^2}{2g} = 0. \tag{14}$$

Between (2) and (4), assuming a short length of line, and with a discharge valve coefficient K_P in terms of the velocity V_2,

$$\frac{P_2 - P_4}{w} + z_2 - z_4 + \frac{V_2^2}{2g} - \frac{V_4^2}{2g} - K_P\frac{V_2^2}{2g} = 0 \tag{15}$$

Adding equations (13), (14), and (15), noting that $p_0 = 0$, $V_0 = 0$, and $V_1 = V_2 = V$,

$$-\frac{p_4}{w} + z_0 - z_4 - \frac{V^2}{2g}\left(f\frac{L}{D} + K_e + K_p\right) - \frac{V_4^2}{2g} = \frac{L}{g}\frac{dV}{dt} \tag{16}$$

V_4 in equation (16) is the velocity existing from the discharge valve and can be expressed in terms of the drive line velocity.

$$A_4V_4 = AV \tag{17}$$

Equation (16) then becomes,

$$-\frac{p_4}{w} + z_0 - z_4 - \left(\frac{A^2}{A_4^2} + f\frac{L}{D} + K_e + K_p\right)\frac{V^2}{2g} = \frac{L}{g}\frac{dV}{dt} \tag{18}$$

From equation (3), $h = \frac{p_4}{w} + z_4 - z_0$

Defining

$$N = \frac{A^2}{A_4^2} + f\frac{L}{D} + K_e + K_p \tag{19}$$

equation (18) is then

$$-h - N\frac{V^2}{2g} = \frac{L}{g}\frac{dV}{dt} \tag{20}$$

Equations (12) and (20) can be solved for the time intervals of the drive flow and of the discharge flow.

$$\text{Drive interval} = t_c - t_b = \left(\frac{2L^2}{gHM}\right)^{1/2}\tanh^{-1}\left(\frac{MV_m^2}{2gH}\right)^{1/2} \tag{21}$$

$$\text{Discharge interval} = t_f - t_c = \left(\frac{2L^2}{ghN}\right)^{1/2}\tan^{-1}\left(\frac{NV_m^2}{2gH}\right)^{1/2} \tag{22}$$

In equations (21) and (22), V_m is the maximum velocity in the drive pipe which occurs when the drive valve closes and the discharge valve opens. The flow volume in each time interval is obtained from equations (21) and (22).

$$(\mathrm{Vol})_T = \int_{t_b}^{t_c} AV\,dt = \int_0^{V_m}\frac{AV}{g}\,\frac{L\,dV}{\left(H - M\frac{V^2}{2g}\right)} \tag{23}$$

$$(\mathrm{Vol})_P = \int_{t_c}^{t_f} AV\,dt = \int_{V_m}^{0} \frac{AV}{g} \frac{L\,dV}{\left(h + N\dfrac{V^2}{2g}\right)} \tag{24}$$

Integrating equations (23) and (24)

$$(\mathrm{Vol})_T = \frac{LA}{M}\left[\log\left(\frac{1}{1 - \dfrac{MV_m^2}{2gH}}\right)\right] \tag{25}$$

$$(\mathrm{Vol})_P = \left[\frac{LA}{N}\log\left(\frac{NV_m^2}{2gh} + 1\right)\right] \tag{26}$$

The volume ratio per cycle is also the flow rate ratio since the volumes may be averaged over the total time of a cycle.

$$\frac{Q_P}{Q_T} = \frac{(\mathrm{Vol})_P}{(\mathrm{Vol})_T} = \frac{M\log\left(\dfrac{NV_m^2}{2gh} + 1\right)}{N\log\left(\dfrac{1}{1 - \dfrac{MV_m^2}{2qH}}\right)} \tag{27}$$

If the drive valve was kept open until the drive line velocity reached the steady flow valve under the head, H, this steady flow velocity would be obtained from equation (12) with $dV/dt = 0$, and $V_s^2 = 2gH/M$. With this definition equation (27) can be changed to the form,

$$\frac{Q_P}{Q_T} = \frac{M}{N}\frac{\log\left(\dfrac{V_m^2}{V_s^2}\dfrac{N}{M}\dfrac{H}{h} + 1\right)}{\log\left(\dfrac{1}{1 - \dfrac{V_m^2}{V_s^2}}\right)} \tag{28}$$

The efficiency is

$$e = \frac{Q_P\,h}{Q_T\,H} = \left(\frac{h}{H}\right)\left(\frac{M}{N}\right)\frac{\log\left(\dfrac{V_m^2}{V_s^2}\dfrac{N}{M}\dfrac{H}{h} + 1\right)}{\log\left(\dfrac{1}{1 - \dfrac{V_m^2}{V_s^2}}\right)} \tag{29}$$

The pumped flow rate is the pumped volume in a cycle averaged over the total time interval of a cycle.

$$Q_P = \frac{(\mathrm{Vol})_P}{t_f - t_b} = \frac{A\log\left(\dfrac{V_m^2}{V_s^2}\dfrac{N}{M}\dfrac{H}{h} + 1\right)}{\left(\dfrac{2M}{gh}\right)^{1/2}\left[\left(\dfrac{N}{M}\right)\tan^{-1}\left(\dfrac{N}{M}\dfrac{H}{h}\dfrac{V_m^2}{V_s^2}\right) + \left(\dfrac{h}{H}\right)^{1/2}\tanh^{-1}\left(\dfrac{V_m}{V_s}\right)\right]} \tag{30}$$

Equations (28), (29), and (30) give the complete performance information for the hydraulic ram.

The performance equations contain three ratios, V_m/V_s, N/M, and H/h. These may be reduced to two ratios, noting that M and N are approximately equal,

$$M = \left(\frac{A}{A_3}\right)^2 + f\frac{L}{D} + K_e + K_d$$

$$N = \left(\frac{A}{A_4}\right)^2 + f\frac{L}{D} + K_e + K_p$$

Note that $K_e \simeq 0.5$, K_p and K_d are $\simeq 1.0$, A_3 and A_4 are approximately the same areas. Also fL/D is the same for both M and N with the previous assumption that f is not dependent on the velocity, and has been taken as a constant. For minimum values of M and N, A/A_3, A/A_4, K_p, and K_d should be as small as possible. If these approach zero, or are small compared to

$$f\frac{L}{D} + K_e, \text{ then } M = N \text{ and } M/N = 1.$$

The performance can then be reduced to

$$\frac{Q_P}{Q_T} = \frac{\log\left(\dfrac{V_m^2}{V_s^2}\dfrac{H}{h} + 1\right)}{\log\left(\dfrac{1}{1 - \dfrac{V_m^2}{V_s^2}}\right)} \tag{31}$$

$$e = \frac{h\log\left(\dfrac{V_m^2}{V_s^2}\dfrac{H}{h} + 1\right)}{H\log\left(\dfrac{1}{1 - \dfrac{V_m^2}{V_s^2}}\right)} \tag{32}$$

$$\frac{Q_P}{AV_s} = (1/2)\left(\frac{h}{H}\right)^{1/2} \times \frac{\log\left(\dfrac{V_m^2}{V_s}\dfrac{H}{h} + 1\right)}{\left[\tan^{-1}\left(\dfrac{H}{h}\dfrac{V_m^2}{V_s^2}\right)^{1/2} + \left(\dfrac{h}{H}\right)^{1/2}\tanh^{-1}\left(\dfrac{V_m}{V_s}\right)\right]} \tag{33}$$

Using $\dfrac{V_m}{V_s}$ as a parameter, the expected performance is obtained in generalized form as shown in the Figs. 4, 5, and 6. Of interest also is the cycle frequency. With $M = N$,

$$\text{cycles/s} = \frac{1}{t_f - t_b} = \frac{gH}{LV_s}\left(\frac{h}{H}\right)^{1/2} \times \frac{1}{\left[\tan^{-1}\left(\dfrac{V_m^2}{V_s^2}\dfrac{H}{h}\right)^{1/2} + \left(\dfrac{h}{H}\right)^{1/2}\tanh^{-1}\left(\dfrac{V_m}{V_s}\right)\right]} \tag{34}$$

Test Results

Measured performance of a 2-in. commercial ram are reported by O'Brien and Gosline [5]. Their results for two different discharge valve settings are compared to the approximate analysis in Figs. 7, 8, and 9. From Fig. 7 it is seen that the ram head versus discharge data are very well predicted by this approximate analysis. From Figs. 8 and 9 it is seen that the ram ef-

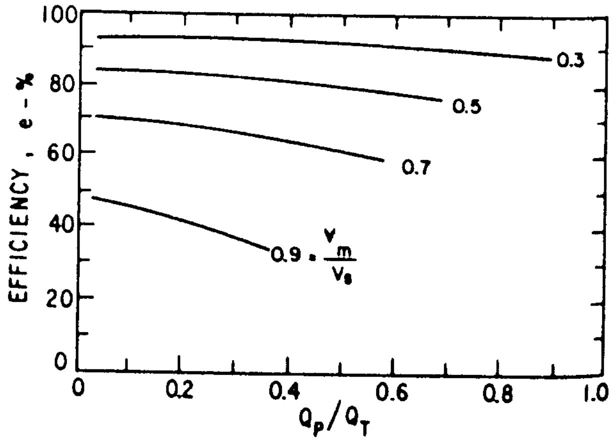

Fig. 4 Generalized analytical performance of the hydraulic ram

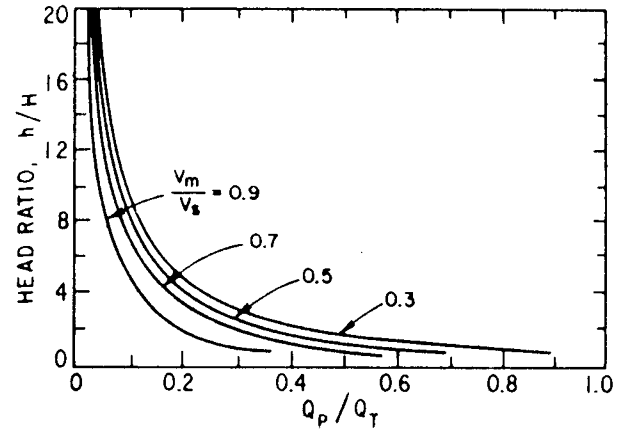

Fig. 5 Generalized analytical performance of the hydraulic ram

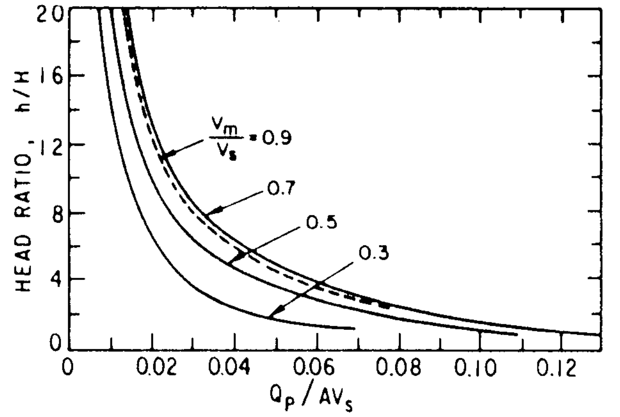

Fig. 6 Generalized analytical performance of the hydraulic ram

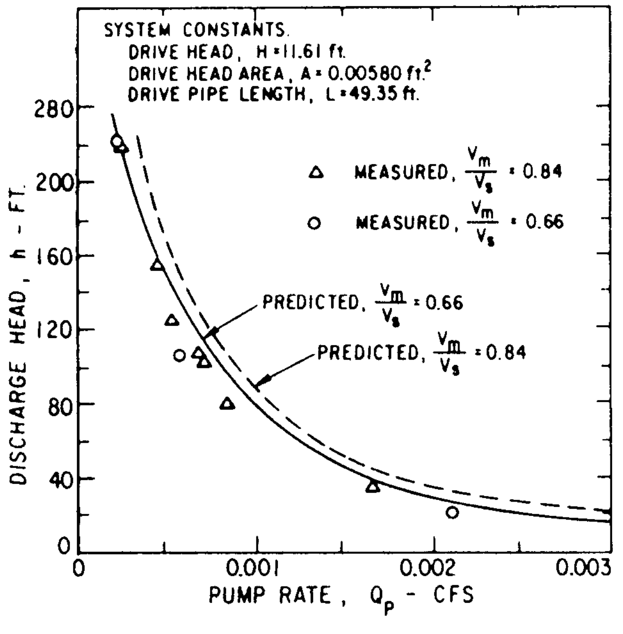

Fig. 7 Predicted and measured performance of a 2-in. ram

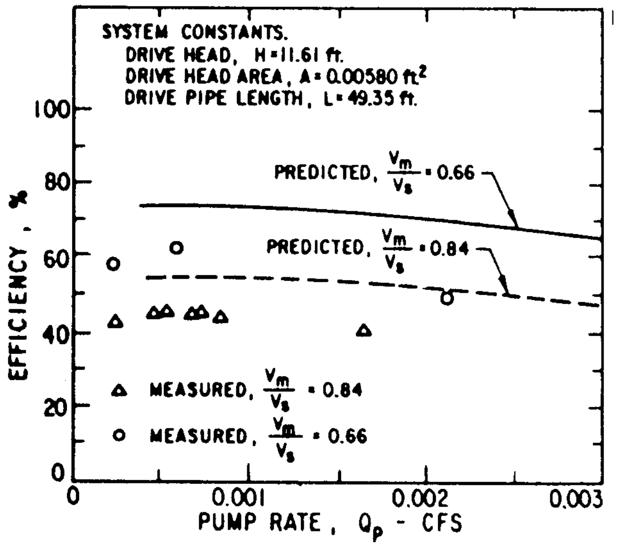

Fig. 8 Predicted and measured performance of a 2-in. ram

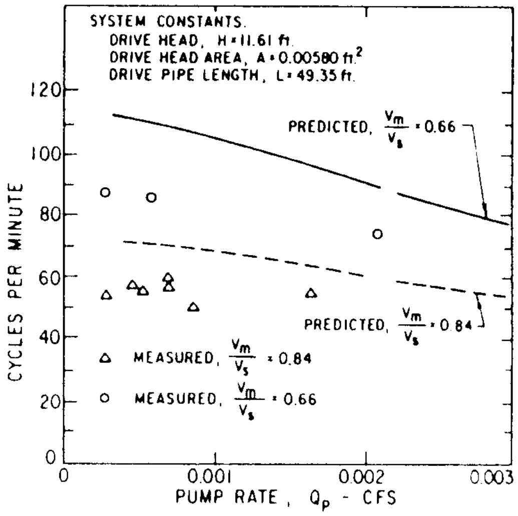

Fig. 9 Predicted and measured performance of a 2-in. ram

ficiency and cyclic frequency versus discharge data are less well predicted, though the trends are in agreement.

The results of a 12-in. ram are reported by Mead [6] with a supply head, H, of 48 ft. However, the steady state velocity was not given so a direct comparison of Mead's data cannot be made for Figs. 7, 8, and 9. Nevertheless, the efficiencies reported are of interest. Mead's data for a 12-in. ram are as follows.

Pump rate Q_P–cfs	Drive rate Q_T–cfs	Pump head h–ft	Eff e	cycles per min
0.555	1.08	82	86.5	65
0.755	1.53	83	85.6	50
0.846	1.78	85	85.0	45
0.915	1.91	83	83.3	41
0.930	2.04	85	81.8	40
1.020	2.04	79	83.4	37
1.080	2.06	87	74.8	32

Other test results on a 10-in. ram show efficiencies of 90 percent (Mead [6]).

If the best rotary turbine connected to the best rotary pump was to be designed for the same service the combined efficiency would be *less* than that of the hydraulic ram. For example, a turbine with an efficiency of 90 percent driving a pump with an efficiency of 90 percent would have an overall efficiency of 81 percent—lower than reported ram efficiencies.

Operational Features

The analysis and the trends of the test results indicate that best performance is obtained when the drive line velocity at closure of the drive valve is as small as possible consistent with continuous reliable operation of the ram. The drive valve, and the discharge valve, should be so designed that closures and openings are rapid. Since stoppage is against the valve seat on closure, or against a retainer on opening, these surfaces should be cushioned with hard rubber or other suitable material to absorb the stoppage shocks.

The analysis does not reveal the optimum length of drive line. On the basis of tests of a 1/2-in. ram and a 1-1/2-in. ram, Calvert [1, 2], found that optimum performance for all head ratios was obtained with

$$150 < L/D < 1000.$$

The cyclic frequency of operation decreased as the drive line length increased with the product of the frequency and length roughly constant. Calvert states, "Practical aspects such as valve wear, liability to fatigue and toleration of the noise all suggest a low frequency of beat and hence a longer drive pipe than the minimum for good performance."

The surge tank into which the pumped fluid discharges before delivery to the pump unit discharge line is a necessary feature of the ram. Kroll [3] states that the air volume should be approximately 100 times the volume of water delivered per cycle.

In order to keep the surge tank charged with air a snifter valve is used. Recall that at the end of the pumping portion of the cycle the discharge valve closes when the velocity in the discharge, and in the drive line, becomes zero and reverses. On complete closure of the discharge valve the pressure in the pipe leading to the discharge valve becomes less than atmospheric momentarily until the drive valve opens. A small check valve (snifter valve) in the pipe leading to the discharge valve can then be arranged to admit a small amount of air each cycle to replenish the air that goes into solution or mixes with the final liquid discharge from the surge tank.

A ram is normally installed to run continuously, feeding into a discharge reservoir which is the source for utilization of the pumped fluid. The discharge reservoir acts as a balance with variable demand from the reservoir. If the demand is less than the supply over a period of time such that the reservoir fills, excess supply is allowed to overflow and is carried off to any convenient drainage system. This may be considered inefficient operation, but since no energy supply is used other than the continuously available low head of the supply reservoir which would be wasted anyway in its normal course down the stream bed, there is no net energy cost.

Continuous operation of the ram is necessary unless elaborate starting mechanisms are incorporated for intermittant operation. Reasonable performance dictates a drive valve design which results in a valve size and weight with which the valve will be kept closed by the hydrostatic head of the supply. Initial starting is done manually, by hand on small rams or by some mechanical linkage on large rams. After a few manual actuations of the drive valve the pressure increase in the discharge surge tank will take over to initiate and continue the automatic cycling operation.

Conclusions

The simple mechanism and application of the hydraulic ram shows it to be a unique device which delivers pumped flow to a high head from an available low head flow, as for example in a pump system using the head drop in a stream to deliver flow uphill for domestic and irrigation supply. No other pump system uses the available energy without electrical or other mechanical devices. In addition, the hydraulic ram, in certain applications, is as good as or better than other pump systems is simplicity and performance. An approximate analysis is shown to give performance features of heads and flow rates which relate directly to measured performances.

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The Automatic Hydraulic Ram

By J. Krol, Ph.D., A.M.I.Mech.E.*

After describing the operation of a typical hydraulic ram installation, the paper reviews the fundamentals of the water hammer as a prerequisite to the proper understanding of the limitations of this hydraulic machine. The historical development is discussed in some detail with the object of ascertaining what research work remained to be done. The author presents his own theory based on the application of general laws of mechanics to the study of a specially designed experimental hydraulic ram. By means of a theory developed, which agrees satisfactorily with experiment, it is possible to forecast the behaviour of any automatic hydraulic ram, provided that the following four properties of a given installation have been determined separately by experiment: (*a*) loss of head in the drive pipe; (*b*) loss of head due to the impulse valve; (*c*) drag coefficient of the impulse valve; and (*d*) head lost during the period of retardation.

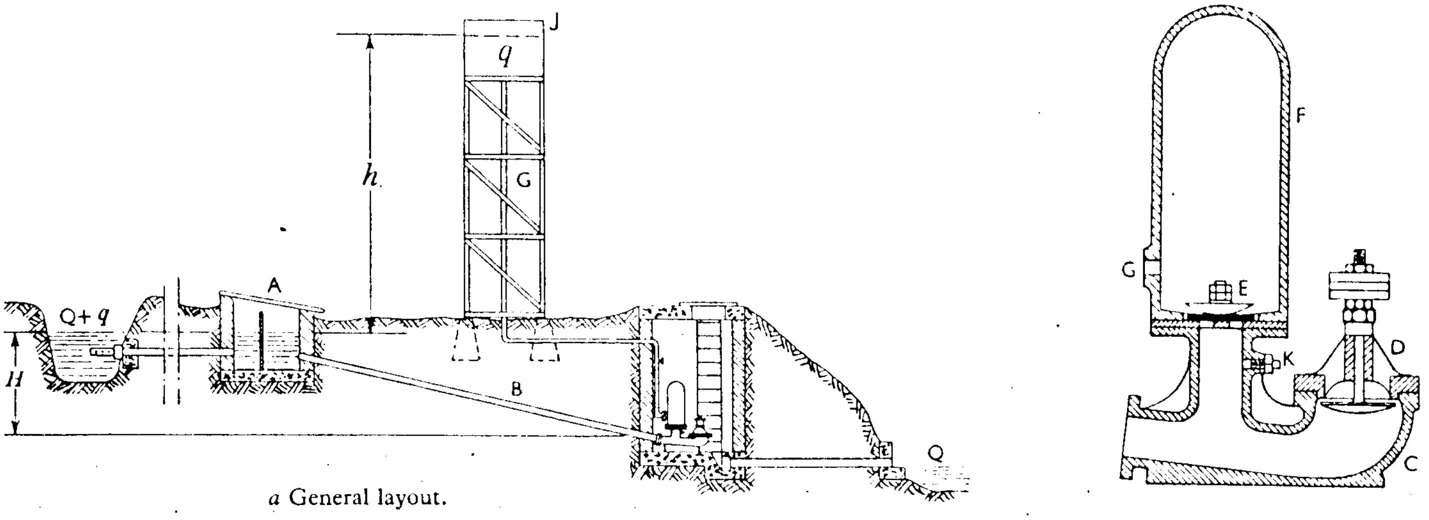
INTRODUCTION

Although the automatic hydraulic ram was invented about 175 years ago, its use had not become as widespread as its simplicity of construction, low cost, dependability, ease of operation, and efficiency would seem to warrant. This has been due largely to the lack of reliable information concerning the limiting conditions under which the ram is applicable and the phenomena governing its action.

The automatic hydraulic ram may be defined as a self-operating water raising machine, combining the qualities of prime-mover and pump, by which the fall of a comparatively large volume of water under a comparatively low head provides power to raise a portion of a supply water to a height greater than that of the source, by taking advantage of the pressure rise resulting from the abrupt interruption of the motion of a column of water.

In Fig. 16, which illustrates a typical hydraulic ram installation, a supply tank, A, is fed from a spring, stream, river, or other source of water. An inclined conduit, B, called the drive pipe, is the essential part of the machine in which the potential energy of the supply water, *QH*, is first converted into kinetic energy and subsequently into the potential energy of water delivered, *qh*. The kinetic energy is obtained by a quantity of water, Q, falling under a head, *H*, from the tank, A, through the drive pipe, B, then passing through the impact chamber, C, and finally escaping by the impulse valve, D. The velocity of flow increases under the influence of the supply head until the dynamic pressure acting on the underside of the impulse valve, D, becomes great enough to overcome its weight. The impulse valve closes rapidly and, owing to water hammer, the pressure inside the drive pipe rises sufficiently to force open the delivery valve, E, thus discharging through it a quantity of water, *q*. The air chamber, F, connected with the delivery pipe, G, prevents the occurrence of the water hammer in the delivery pipe and reduces the fluctuations in the rate of discharge to the tank, J, from which the water is distributed by gravitation, as required. As soon as the momentum of the supply column is exhausted, the delivery valve, E, closes, and the water contained in the drive pipe recoils towards the supply tank, A. This recoil removes the pressure acting on the underside of the impulse valve, D, which thereupon falls and again allows the escape of water. Simultaneously, the recoil causes the air-charging valve, K, to open, admitting a small amount of air into the impact chamber of the ram. This air is carried along with the water into the air chamber to compensate for that absorbed by the water.

The above cycle, consisting of two main periods, namely, the period of acceleration during which the kinetic energy is produced, and the period of retardation during which the water is pumped, is automatically repeated indefinitely at a rate varying from a few to over 300 pulsations a minute.



a General layout.

b Elevator detail.

Fig. 16. Typical Hydraulic Ram Installation

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Notation.

A	Projected area of impulse valve seat, sq. ft.
C	Velocity of acoustic wave in an elastic medium, ft. per sec.
D	Diameter of drive pipe (internal), feet.
E	Modulus of elasticity of drive pipe material, lb. per sq. in.
f	Friction coefficient of drive pipe.
g	Acceleration due to gravity, 32·2 ft. per sec. per sec.
h	Delivery head measured above the level of supply water, feet.
h_c	Level of water in the air chamber measured over the base of impulse valve, feet.
h_r	Head lost during the retardation period in the drive pipe, delivery valve, and delivery pipe, feet.
h_{max}	Maximum delivery head developed by the ram, feet.
H	Static supply head measured above the impulse valve, feet.
K	Bulk modulus of elasticity of water, lb. per sq. in. (approximately 300,000 lb. per sq. in.).
K_1	Virtual bulk modulus of elasticity of water and pipe, lb. per sq. in.
L	Length of drive pipe measured from supply tank to impulse valve, feet.
P_d	Power developed, ft.-lb. per sec.
p	Pressure rise due to water hammer, lb. per sq. in.
$1/m$	Poisson's ratio.
q	Quantity of water delivered per cycle, lb.
q_c	Part of the quantity q temporarily stored in the air chamber, lb.
Q	Quantity of water flowing to waste per cycle, lb.
Q_2	Quantity of water flowing to waste during period 2, lb.
Q_3	Quantity of water flowing to waste during period 3, lb.
$R(s)$	Coefficient of loss of head due to impulse valve.
r	Radius of pipe, feet.
s	Length of impulse valve stroke, inches.
t	Time in general, seconds.
t_n	Duration of period n (where $n = 1, 2, 3, 4, 5, 6, 7$), seconds.
T	Duration of a complete cycle, seconds.
t_p	Thickness of drive pipe walls, feet.
v	Velocity in general, ft. per sec.
v_n	Velocity of water in the drive pipe at the end of period n (where $n = 1, 2, 3, 4, 5, 6, 7$), ft. per sec.
w	Specific weight of water, 62·4 lb. per cu. ft.
W	Weight of impulse valve, lb.
y	Distance travelled by water column in the drive pipe due to recoil, feet.
Δ^v	Reduction in velocity causing water hammer, ft. per sec.
Σj	Sum of coefficients of turbulence losses due to bends and obstacles in the drive pipe.
η	Efficiency.
$\Phi(s)$	Drag coefficient of impulse valve.

WATER HAMMER

The phenomena of the water hammer have been studied in detail by a number of investigators (A.S.M.E. 1933)*. However, to understand correctly the action of the automatic hydraulic ram it is necessary to review the basic formulae for the pressure rise through the water hammer in a rigid pipe and an elastic pipe respectively.

If a column of water moving with a steady velocity through a rigid pipe of uniform diameter has its motion instantaneously retarded by the partial or complete closure of a rigid valve, the phenomena experienced are due to the elasticity of the column of water alone, and are analogous to those obtaining in the case of the longitudinal impact of an elastic bar against a rigid wall. Mathematically, the excess pressure due to the water hammer in a rigid pipe can be expressed by the formula

$$p = (\Delta^v/12)\sqrt{wK/g} \quad . \quad . \quad . \quad . \quad . \quad (33)$$

Substituting these data in equation (33), it will be found that an instantaneous reduction in velocity of 1 ft. per sec. is equivalent to the pressure head of 63·3 lb. per sq. in., or 146 feet of water.

For water in a drive pipe, owing to the elasticity of the pipe material, part of the kinetic energy of the moving column of water is expended in straining the walls of the pipe both longitudinally and circumferentially. The effect is the reduction of the value of K, obtaining with a rigid pipe, to a virtual value of K_1, for an elastic pipe. The equations for the determination of K_1 were developed by Joukowski (1898) and Gibson (1908).

When the pipe is considered to be fixed at the ends and free to expand radially, Joukowski arrived at the equation

$$K_1 = 1\Big/\left(\frac{1}{K}+\frac{2r}{t_pE}\right) \quad . \quad . \quad . \quad . \quad (34)$$

Professor Gibson, on the other hand, assumed that the pipe was free to expand radially and longitudinally, and reached the equation

$$K_1 = 1\Big/\frac{1}{K}+\frac{r}{2t_pE}\left(5-\frac{4}{m}\right) \quad . \quad . \quad . \quad (35)$$

Fig. 17 shows the numerical values of maximum pressure rise for pipes made of steel, wrought iron, cast iron, copper, lead, and concrete, computed as a function of r/t_p with the aid of equation (35). Data for the values of Poisson's ratio and modulus of elasticity for these materials were based on information given by Walker and Crocker (1939).

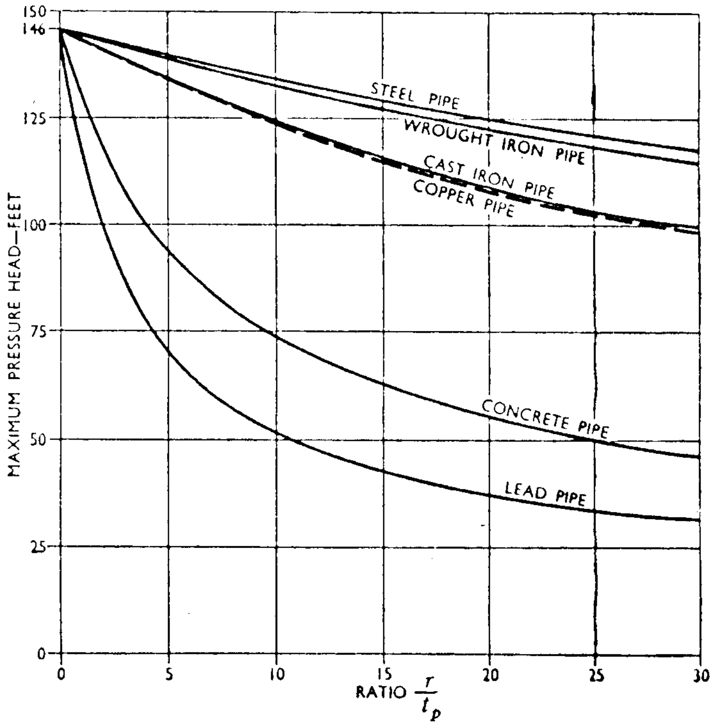

Fig. 17. Variation of Maximum Pressure Head Due to Instantaneous Reduction in Velocity of 1 ft. per sec. with the Ratio r/t_p

The analysis of equations (33), (34), and (35) enables some conclusions to be drawn in regard to the suitability of various engineering materials for the drive pipe. From Fig. 17 a steel pipe secures the highest pressure head; wrought iron and cast iron come next. Actual practice confirms this conclusion, as steel, wrought iron, and cast iron are the materials most commonly used for the construction of drive pipes. Further, other conditions being equal, the material of the pipe and its ratio r/t_p will affect the performance of the hydraulic ram. Eytelwein (1805) used a copper drive pipe in his experimental ram, and his results would have been much better if he had used a steel drive pipe of the same dimensions. Clark (1900) mentions the possibility of lead being used as a material for drive pipes, but from Fig. 17 lead is the most unsuitable material for that purpose. Grandemange (1933) suggested concrete drive pipes for rams dealing with very large quantities of water. From Fig. 17 such drive pipes should be very thick to produce a water hammer comparable with that in a steel pipe of the same diameter.

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* An alphabetical list of references is given in the Appendix, p. 64.

PRACTICAL APPLICATIONS

If simplicity were the only factor involved, hydraulic rams would be in universal use, for, from their construction, they are the simplest known automatic pumping units. But, unfortunately, their application is restricted by certain material topographic features. For use of an hydraulic ram, two primary conditions are required—a steady and sufficient supply of water, and a fall in the contour of the land so that the necessary driving power may be obtained and the escaping water can be promptly disposed of.

The size of the hydraulic ram is usually denoted by the nominal diameter of the drive pipe, which depends upon the available quantity of water. The range of sizes in general use is as follows: ¾ inch, 1–4½ inches in ½-inch increments, and 5–12 inches in 1-inch increments. Hydraulic ram manufacturers in Britain prefer drive pipes of lower diameter, generally not exceeding 6 inches. Larger installations up to 12 inches diameter are used in the United States, and there is no reason why 12 inches should constitute a limit to the diameter of the drive pipe. Grandemange suggested concrete drive pipes with diameters of a few feet.

The minimum flow of water with which a small ram of diameter ¾–1 inch will operate is approximately 3 gal. per min. For rams of 12 inches diameter a minimum of 350 gal. per min. is common practice. Rams designed for dealing with very large supplies of water have been made to use 700 gal. per min., and in high supply heads, the figure may become as much as 1,000 gal. per min. When used in batteries, hydraulic rams may utilize any amount of water.

It is generally desirable that the minimum difference in levels of the source of water supply and the impulse valve should be some 3 feet, although where drive pipes of larger diameters, say, above 6 inches, are employed the difference may be as little as 2 feet. These limits are not rigid, as hydraulic rams have been operated with a head of only 1 inch. The maximum supply head is theoretically unlimited, but in actual practice seldom exceeds 30 feet. Ringelmann (1908) examined technical data relating to 993 installations, and found that the supply heads most utilized were those between 3 and 6 feet, next between 6 and 12 feet, and last between 12 and 30 feet. The late Professor Mead (1933) designed and successfully operated an experimental ram under a supply head of 100 feet.

The delivery head is also subject to some limitations. If the delivery head is very low, the hydraulic ram will not operate satisfactorily, because the resulting recoil is inadequate to open the impulse valve. The minimum delivery head is approximately twice that of the supply head, and for this reason, under most favourable conditions, the ram cannot deliver more than half the water flowing through the drive pipe. An insufficient delivery head can be increased by throttling the valve controlling the delivery pipe. The maximum delivery head depends principally upon the velocity of the water at the moment of impulse valve closure. Since an instantaneous reduction of 1 ft. per sec. of velocity of water produces the pressure head at about 146 feet, it can be seen that a hydraulic ram may pump water against a very considerable head. According to Cleverdon (1937), standard American rams are guaranteed to pump against a head of 250 feet. Rams specially designed may pump against a head within the limits of 600–1,000 feet. These rams are constructed heavily and of great strength to resist the severe water hammer. Cleverdon points out that a delivery head equal to thirty times the available supply head is a satisfactory maximum for most rams.

The horizontal distance to which the water can be forced should be expressed in terms of an additional head of water added to the required static delivery head. Numerous hydraulic rams operate with delivery pipes up to 4 miles long.

Durability and dependability of a typical hydraulic ram installation are very satisfactory. Anderson (1922) reports a case of a ram which had been at work for over 100 years. In another ram the valves had not been inspected for twenty-seven years. Consequently, the cost of maintenance is negligible, since there are no parts which require lubrication and no attention is needed other than periodical inspection.

The efficiency of well constructed and adjusted hydraulic rams is excellent. The Maple Leaf pumping station for the City of Seattle, United States of America, consists of two hydraulic rams of 12 inches diameter, and has a range of capacity of water pumped from 720,000 to 1,300,000 gallons per day at 91 per cent efficiency (Carver 1918).

Perhaps the only disadvantage of the hydraulic ram installation is a certain amount of noise. Insertion of a piece of rubber tube between the body of a ram and the delivery pipe is claimed by Kirchoffer (1916) to mitigate this nuisance.

The requirements for operating hydraulic rams make them especially adaptable to ensure a continuous delivery of water for:—

(*a*) rural communities, such as farms, country houses, hotels, and factories;
(*b*) irrigation schemes;
(*c*) isolated locomotive water towers; and
(*d*) villages and small towns.

PAST DEVELOPMENT

The discovery of the underlying principle of the hydraulic ram and its first practical application are due to John Whitehurst, F.R.S., of Derby, who built his apparatus about 1772. Whitehurst erected his machine at Oulton, Cheshire, for use at the local brewery. His invention (Fig. 18) consisted of an inclined pipe, A, 600 feet long and 1½ inches diameter, connecting a supply tank, T, with an air chamber, D, controlled by a check valve, C. A pipe, B, was branched from the main near its lower end and fitted with a hand operated tap, E, located 16 feet below the level of water in the supply tank. A child was employed to operate the tap, E, and with each closure the phenomenon of a water hammer was caused, driving a portion of the supply water to the delivery tank, F. Although the Whitehurst apparatus was exceedingly crude, it was, nevertheless, actually used until about 1800. Whitehurst (1775) called public attention to his invention through a communication to the Royal Society.

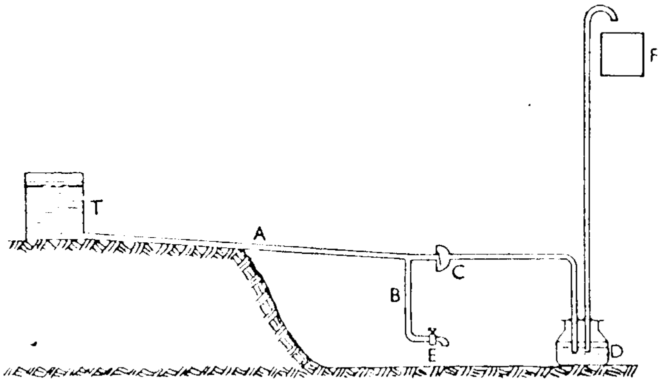


Fig. 18. Whitehurst's Hydraulic Machine

In 1776, Joseph Michael de Montgolfier, a French paper manufacturer and co-inventor of the first hot-air balloon, invented a machine based on the same fundamental principle, but acting automatically and thus eliminating the human intervention required for the operation of Whitehurst's apparatus. The arrangement of the original Montgolfier's ram is shown in Fig. 19. Its construction has all the important features of the typical hydraulic ram described in connexion with Fig. 16. Montgolfier named his machine "*le belier hydraulique*", from which the term "hydraulic ram" was derived and is universally accepted.

After Montgolfier's death the development of the hydraulic ram followed two main directions, namely, improvements in mechanical design, and research into principles of its action.

Improvements in mechanical design can be subdivided into two sections: (1) development of the ordinary type of hydraulic ram, similar to that shown in Fig. 16, and (2) development of its variations. In the first section, remarkable designs were originated by the inventor's son François Montgolfier and many others.

In the second section, variations of the hydraulic ram, possessing distinctive features of their own, are:—

(*a*) Combination with a siphon.
(*b*) Combination with a suction pump.
(*c*) Action as an air compressor.
(*d*) With a mechanically operated impulse valve.
(*e*) Adapted as a motor.
(*f*) Adapted to the artesian well.
(*g*) Without valves or hydro-pulsator.
(*h*) Constructed of reinforced concrete.
(*i*) Adapted for utilizing tidal power.

Previous investigations to find an explanation for the behaviour of the automatic hydraulic ram may be classified under three headings: empirical, theoretical, and rational.

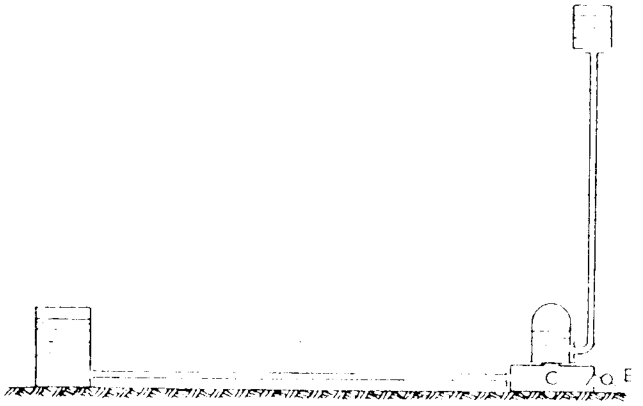


Fig. 19. Montgolfier's First Hydraulic Ram

The empirical method, based on experimental tests but not correlated with theory, resulted in numerous rules of thumb, which were misleading since the action of an hydraulic ram depends upon so many variables. Among the empirical researchers were Eytelwein (1805), d'Aubuisson (1840), and Morin (1863), whose delusive formulae can be still found in engineering reference works. Walker Fyfe (1922), who built many [illegible] installations in the United Kingdom, confirms that he found Eytelwein's and d'Aubuisson's formulae of no [illegible]

In the theoretical method, the problem is attacked by attempting to ascertain the rate of change of the variable velocity of the column of water during each period of the working cycle. From this relation, the quantities of water escaping and delivered during each cycle may be found and the duration of the latter determined. The solution of the problem by theoretical methods alone is impossible, as numerous constants relating to the particular type of hydraulic ram can be found only by experiment. Moreover, without facilities for observation of the actual performance, a theoretician is tempted to make assumptions leading to faulty conclusions. Consequently, the examination of past theories proves to be largely a study of failures and approximations. The limitations of past theoretical researches into the action of the automatic hydraulic ram are mainly due to their omission of one or more of the following effects: (*a*) loss of head due to friction and turbulence, (*b*) length of impulse valve stroke, (*c*) load acting on the impulse valve, (*d*) resilience resulting from the elasticity of water and material of the drive pipe, and (*e*) duration of the period during which the impulse valve closes.

The only rational method of research, based on theoretical analysis verified by experiments, is of comparatively recent origin in regard to the automatic hydraulic ram. The first attempt was made by Harza (1908) at the University of Wisconsin. Unfortunately, Harza designed his experimental apparatus with the impulse valve actuated not by water in the drive pipe but by an external motor. The conditions of operation were thus basically changed and his conclusions could not be applied to the automatic hydraulic ram without very serious reservations. However, credit must be given to Harza for experimental confirmation of the laws relating to the unsteady flow of water during the acceleration period.

The next rational investigation is due to O'Brien and Gosline (1933), who carried out their study at the University of California using a 1-inch diameter commercial ram of "Goulds" type. Although the investigation of O'Brien and Gosline provided a first reasonably satisfactory explanation of the action of the automatic hydraulic ram, it has some limitations owing to their application of a commercial ram for experimental study of the problem. Consequently, a number of variables, for example, the load acting on the impulse valve, were inadequately treated. Lansford and Dugan (1941) at the University of Illinois carried out an investigation on two hydraulic rams of "Rife" type, which differed considerably in construction and performance from the one used by O'Brien and Gosline. Although Lansford and Dugan obtained much valuable data for the particular type of hydraulic ram, their investigation has, however, the same limitations as that of O'Brien and Gosline. In particular, neither of the American investigators developed theoretical formulae for the efficiency and power obtainable in the hydraulic ram installation, which would show the effect of all hydraulic constants upon the performance of the machine. Without such formulae no rational design is possible, and to fill this gap, the author carried out the theoretical and experimental investigation (Krol 1947b) reported in condensed form in this paper.

EXPERIMENTAL APPARATUS

In the design of the experimental hydraulic ram means were sought by which each of the following factors could be varied independently: (*a*) delivery head, (*b*) length of stroke of impulse valve, (*c*) weight of impulse valve, (*d*) type of delivery valve, and (*e*) volume of air chamber. To observe simultaneously the motion of the impulse valve and the pressure in the drive pipe as a function of time, a special recording apparatus was constructed.

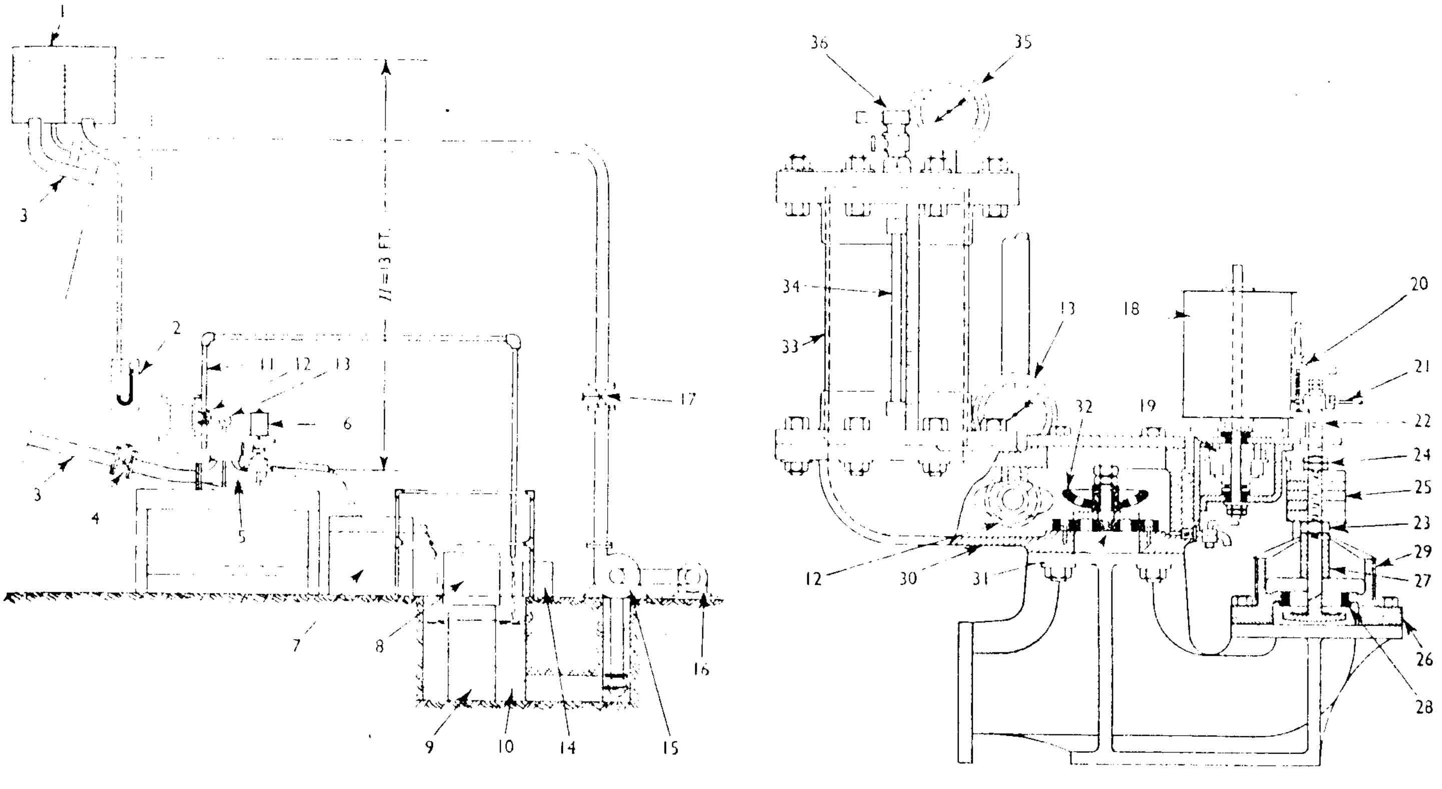
Fig. 20*a* shows the general layout of the experimental installation, which allows for feeding of the supply tank with water collected in the basement tank and simultaneous measurement of the quantity of water delivered and wasted.

Fig. 20*b* shows the detailed elevation of the apparatus. The essential part of the recording apparatus is the drum driven by two small synchronous motors. The gear between the motors and drum shaft is so designed that the speed of a paper band attached to the drum is exactly 100 mm. per sec. A Dobbie-McInnes indicator, connected with the impulse valve box, is attached at a suitable distance from the drum, thus permitting the measurement of pressure–time variations with an error of not more than 0·01 second. Similarly, with the aid of the pencil arm attached to the top of the impulse valve spindle the motion of the latter can be recorded. The pencil arm can slide in its socket and is so adjusted as to secure perfect alignment with the indicator.

To make it as light as possible, the valve spindle was made of Duralumin. The length of the impulse valve stroke can be varied by means of a Duralumin nut and secured in position by a steel lock-nut. To vary the weight of the valve, a series of steel washers ½ inch thick was used. These washers were inserted as required between the nut and the lock-nut. The total weight of the valve spindle, nut, lock-nut, and recording pencil arm was 0·605 lb. Each steel washer weighed 0·880 lb. The total weights of the impulse valves used in experimental investigations were 0·605, 1·485, 2·360, 3·245, 4·125, and 5·005 lb.

The valve box of cast iron is provided with a gun-metal sleeve and bushing. To protect the recording apparatus from splashing, the valve box was enclosed within a brass cover to which a short discharging pipe was attached. Although the body of the experimental apparatus was designed to accommodate any type of delivery valve, actual experiments were limited to one most commonly used (Fig. 20*b*), consisting principally of a seat and a shield holding between them a rubber disk.

The air chamber was made of the standard pipe having an inside diameter of 6 inches. The height of this chamber was 12 inches, and the level of water therein contained could be observed by means of a glass water gauge. On the top of the air chamber a Bourdon gauge was attached. With the aid of the water gauge and Bourdon gauge, it was found that the relation between the volume and the pressure of the air contained in the



a General layout.

b Elevation detail.

Fig. 20. Experimental Apparatus

1 Supply tank.
2 Mercury U-tube gauge.
3 Drive pipe, diameter 2 inches.
4 Gate valve.
5 Experimental hydraulic ram.
6 Recording apparatus.
7 Transit tank.
8 Measuring tank for water wasted.
9 Support for measuring tank.
10 Basement tank.
11 Discharge pipe, diameter $\frac{3}{4}$ inch.
12 Throttling valve.
13 Bourdon gauge.
14 Measuring tank for water delivered.
15 Centrifugal pump.
16 Electric motor.
17 Control valve.
18 Drum.
19 Synchronous motors.
20 Indicator.
21 Pencil arm.
22 Impulse valve spindle.
23 Duralumin nut.
24 Steel lock-nut.
25 Steel washers.
26 Valve box.
27 Sleeve.
28 Bushing.
29 Brass cover.
30 Body.
31 Seat.
32 Shield.
33 Air chamber, diameter 6 inches.
34 Glass water gauge.
35 Bourdon gauge.
36 Air cock.

air chamber was isothermal. In the latter part of the experiments a larger air chamber of the same diameter but having a height of 24 inches was tried. Although the experimental ram was provided with an air charging device, the latter was not used as it was considered undesirable to introduce a factor over which there would be no control. Instead, the air chamber was frequently charged with fresh air by means of an air cock, so that in each series of experiments the volume of air contained was kept constant.

Fig. 21*a* shows a general view of the experimental apparatus fitted with the shorter air chamber, and Fig. 21*b* shows it as fitted with the longer air chamber. Fig. 22*a* and *b* show typical diagrams obtained from the recording apparatus. Both these diagrams show that during the period of retardation the delivery valve opens once only, and that its opening is accompanied by a certain amount of vibration, due to the inertia of the moving parts of the indicator combined with that of the delivery valve itself. Otherwise the pressure during the retardation period is almost constant. The motion of the impulse valve is also shown.

To compare theory with experiment, knowledge of the various resistances of the experimental apparatus is absolutely essential. For the automatic hydraulic ram, the number of experimental factors can be reduced to four only, namely:—

(1) $\left(1+\Sigma j+4f\frac{L}{D}\right)$, expressing the loss of head in the drive pipe excluding the impulse valve, where 1 refers to loss due to velocity head, Σj to loss due to bends, elbows, changes of section, etc., and $4f\frac{L}{D}$ to friction inside the drive pipe.

(2) $R(s)$, denoting loss of head by turbulence due to the impulse valve. This factor can be determined as a function of a length of valve stroke.

(3) $\Phi(s)$, denoting the coefficient of drag in the impulse valve. If the value of this coefficient is known the dynamic force acting on the impulse valve can be determined. Factor $\Phi(s)$ can be expressed also as a function of length of valve stroke.

(4) The head, h_r, lost during the period of retardation. This depends mainly on the loss of head due to the delivery valve, delivery pipe, and, to a lesser extent, the drive pipe.

The results obtained in experiments made to measure these factors can be summarized as follows. Under the conditions of a steady flow, the value of the factor $\left(1+\Sigma j+4f\frac{L}{D}\right)$ was found to be 27·1. The values of the factor $R(s)$ are such that

$$R(s) = \frac{2gH}{v^2}-\left(1+\Sigma j+4f\frac{L}{D}\right) \quad . \quad . \quad . \quad (36)$$

and are shown in Fig. 23. The continuous curve passing through the experimental points can be expressed by the empirical equation

$$R(s) = \frac{2{\cdot}43-1{\cdot}06s+10^{0{\cdot}95-13{\cdot}30s}}{s} \quad . \quad . \quad (37)$$

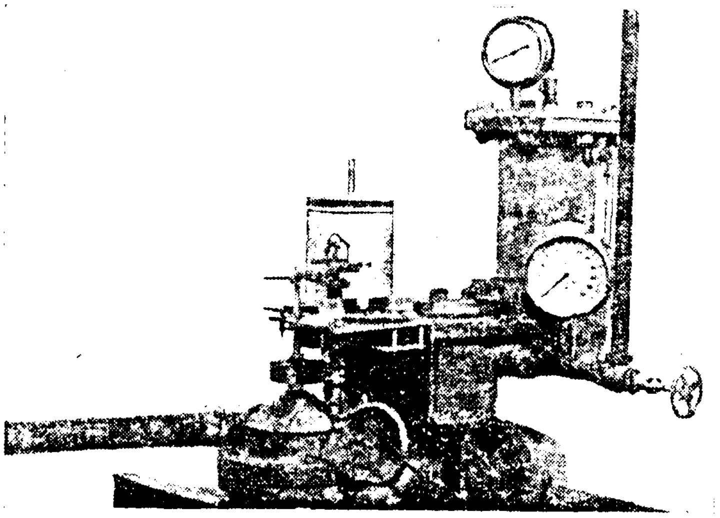
where s is a pure number equivalent to the length of the impulse valve stroke measured in inches. If W is the weight of the impulse valve in pounds, A is its projected area in square feet, and v_2

is the velocity of water at which the impulse valve begins to close, then the values of the factor $\Phi(s)$ are such that

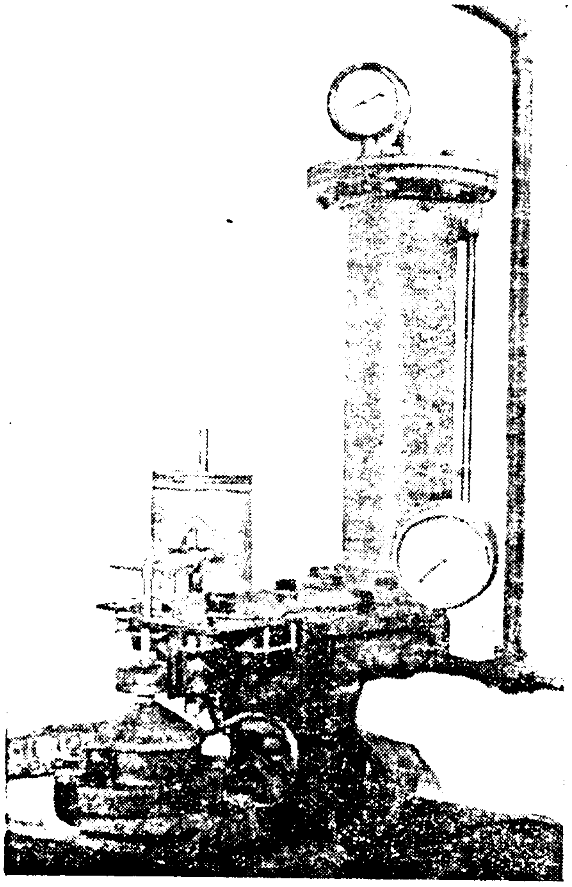
$$\Phi(s) = \frac{Wg}{Av_2^2 w} \quad . \quad . \quad . \quad . \quad . \quad (38)$$

and are shown in Fig. 23. The continuous curve passing through the experimental points can be expressed by the empirical equation

$$\Phi(s) = \frac{0{\cdot}345 - 0{\cdot}275s + 10^{0{\cdot}52 - 6{\cdot}85s}}{s} \quad . \quad . \quad (39)$$

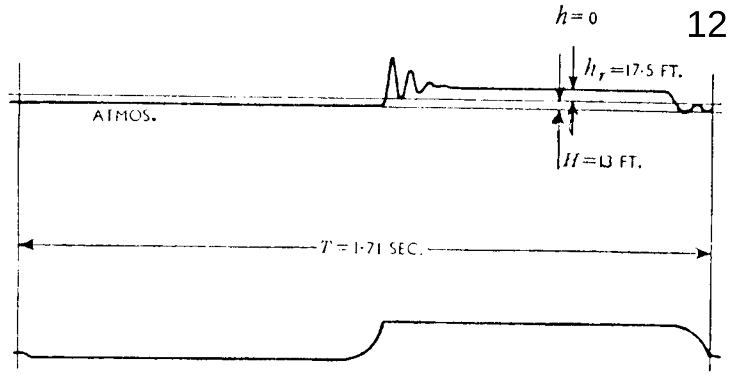


a With short air chamber.

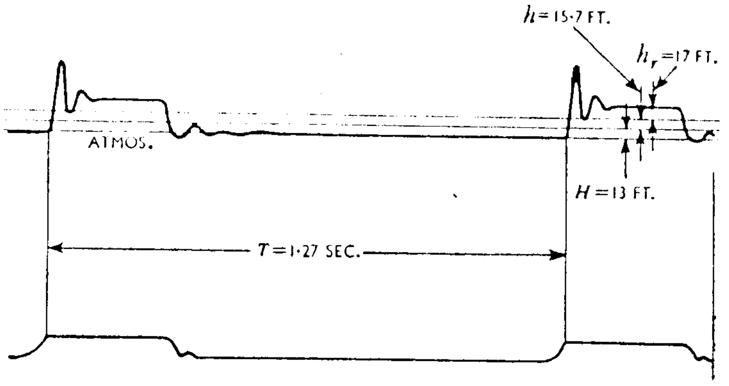


b With longer air chamber.

Fig. 21. Experimental Apparatus



a $s = \frac{5}{16}$ inch; $W = 1{\cdot}485$ lb.



b $s = \frac{3}{16}$ inch; $W = 4{\cdot}125$ lb.

Fig. 22. Typical Recordings

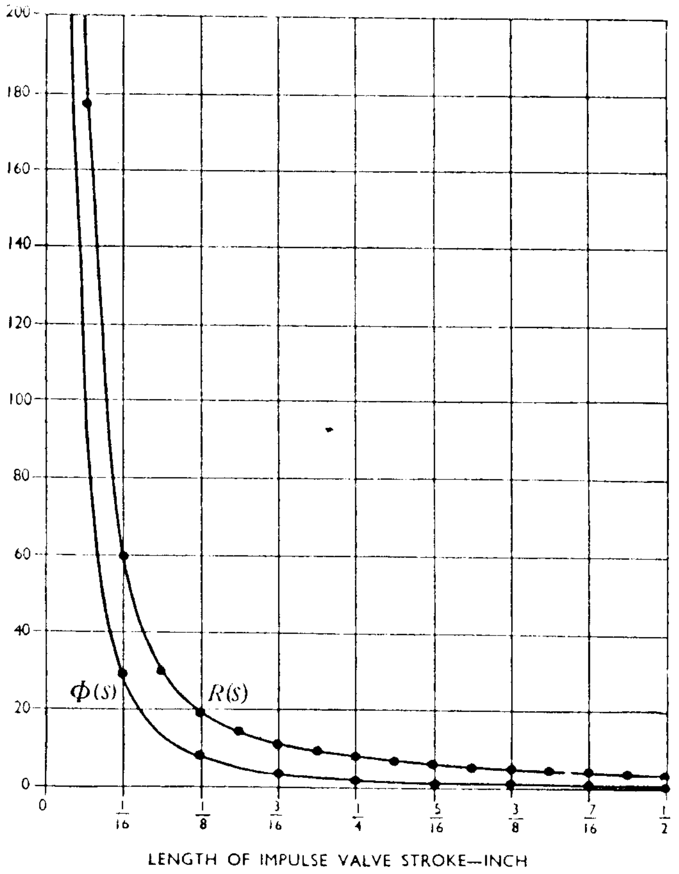


Fig. 23. Coefficients $R(s)$ and $\Phi(s)$ as Functions of Length of Impulse Valve Stroke

● Experimental. ——— Empirical equation.

The head, h_r, lost during the period of retardation can be expressed by the equation

$$h_r = 18{\cdot}5\left(1-\frac{h}{h_{\max}}\right) \quad . \; . \; . \; . \quad (40)$$

where h is the delivery head, and $h_{\max}$ is the maximum pressure head obtainable in the ram. The value of h_r becomes zero when $h = h_{\max}$.

THEORETICAL CYCLE OF OPERATION

The theoretical analysis of the working cycle of the automatic hydraulic ram can best be studied with reference to Fig. 24,

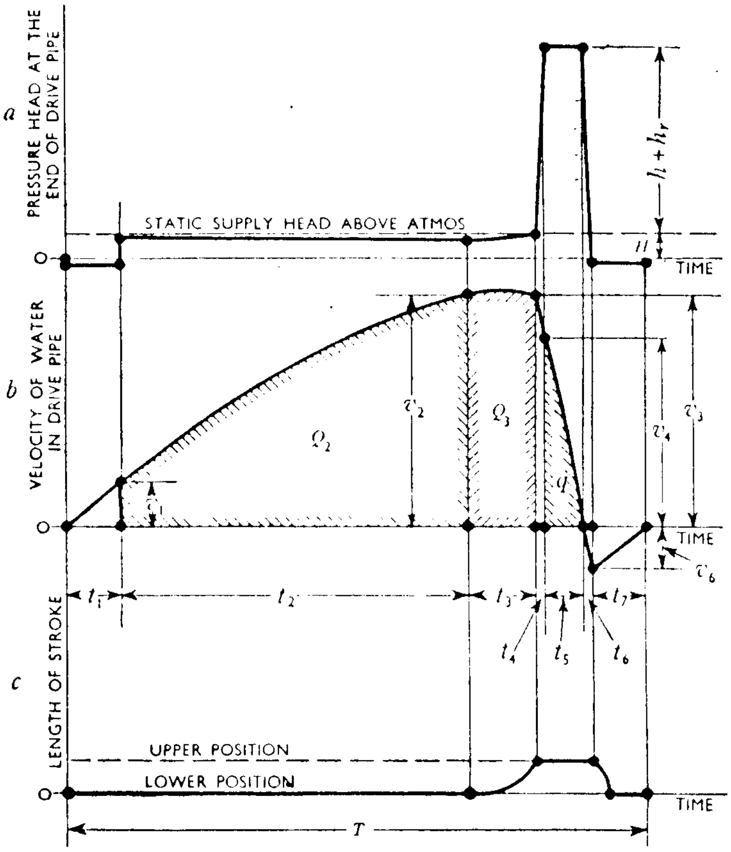


Fig. 24. Graphical Interpretation of the Theory

consisting of three diagrams: a, pressure in the drive pipe as observed in its lower end; b, velocity of water in the drive pipe; and c, simultaneous variation with time of the motion of the impulse valve. For the purposes of mathematical analysis, the total time T of a complete cycle of operation is subdivided into seven periods which are discussed below in more detail.

Analysis of Period 1. The impulse valve is open. Owing to the recoil of water at the end of the preceding cycle, the lower end of the column of water in the drive pipe is separated from the impulse valve by a distance y, the magnitude of which will be determined during the analysis of period 7. Acceleration of water takes place inside the drive pipe, without any water flowing to waste. Theoretically, the water in the drive pipe accelerates according to the equation

$$H-\left(1+\Sigma j+4f\frac{L}{D}\right)\frac{v^2}{2g} = (L/g)\frac{dv}{dt} \quad . \; . \; . \quad (41)$$

which represents the law of the hyperbolic tangent. However, as the values of v are small during period 1, the effect of the second term on the left-hand side of the equation (41) becomes insignificant, and the results obtained are practically identical with those given by the simplified equation

$$H = (L/g)\frac{dv}{dt} \quad . \; . \; . \; . \; . \; . \quad (42)$$

the solution of which gives

$$t_1 = \sqrt{2Ly/gH} \quad . \; . \; . \; . \; . \quad (43)$$

and

$$v_1 = \sqrt{2gHy/L} \quad . \; . \; . \; . \; . \quad (44)$$

Analysis of Period 2. The impulse valve remains open. After returning to the end of the drive pipe with the velocity given by equation (44), the water continues to accelerate until it reaches the velocity

$$v_2 = \sqrt{Wg/Aw\Phi(s)} \quad . \; . \; . \; . \quad (45)$$

at which the impulse valve begins to close. Period 2 usually forms a major portion of the total time of the duration of the complete cycle, T, and frictional resistances must be considered. The equation of motion is therefore

$$H-\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]\frac{v^2}{2g} = (L/g)\frac{dv}{dt} \quad . \quad (46)$$

Integration of the above equation within the limits $v = v_1$ and $v = v_2$ gives

$$t_2 = \frac{L}{\sqrt{2gH\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}}\times$$

$$\log_e\left[\frac{\left(1+\sqrt{\dfrac{W\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}{2AHw\Phi(s)}}\right)}{\left(1-\sqrt{\dfrac{W\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}{2AHw\Phi(s)}}\right)}\times\right.$$

$$\left.\frac{\left(1-\sqrt{\frac{y}{L}\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}\right)}{\left(1+\sqrt{\frac{y}{L}\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}\right)}\right] \quad . \quad (47)$$

and

$$Q_2 = \frac{w\pi D^2L}{4\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}\log_e\left[\frac{\dfrac{2gH}{1+\Sigma j+4f\frac{L}{D}+R(s)}-\dfrac{2gHy}{L}}{\dfrac{2gH}{1+\Sigma j+4f\frac{L}{D}+R(s)}-\dfrac{Wg}{Aw\Phi(s)}}\right]$$

$$. \; . \; . \quad (48)$$

A very important relation can be found by putting

$$2gH/\{1+\Sigma j+4f\frac{L}{D}+R(s)\} = Wg/Aw\Phi(s) \quad . \quad (49)$$

from which

$$W = \frac{2AHw\Phi(s)}{1+\Sigma j+4f\frac{L}{D}+R(s)} \quad . \; . \; . \quad (50)$$

When this relation occurs, the impulse valve will not close at all and Q_2 becomes infinity. It therefore follows that the weight of the impulse valve must satisfy the condition

$$0 < W < \frac{2AHw\Phi(s)}{1+\Sigma j+4f\frac{L}{D}+R(s)} \quad . \; . \; . \quad (51)$$

Equation (50) shows that there is a limitation for the length of the drive pipe L. When this length is such that

$$L \geqslant \frac{[2AHw\Phi(s)-W[1+\Sigma j+R(s)]]D}{4Wf} \quad . \; . \; . \quad (52)$$

the impulse valve will not close at all.

Integrating equation (46) with respect to t within the limits $v = 0$ and $v = v_2$, the duration of both periods 1 and 2 is given by the equation

$$t_1+t_2 = \frac{L}{\sqrt{2gH\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}}\times$$

$$\log_e\left[\frac{1+\sqrt{\dfrac{W\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}{2AHw\Phi(s)}}}{1-\sqrt{\dfrac{W\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}{2AHw\Phi(s)}}}\right] \quad . \quad (53)$$

Analysis of Period 3. Velocity of water is now such that the force acting on the underside of the impulse valve is greater than its weight. The impulse valve begins to close and is completely closed at the end of the period. The rational analysis of period 3 is complicated by the fact that both water and the impulse valve are in motion. However, using the fundamental equation of motion, force = mass × acceleration, and introducing some mathematical simplifications, it is possible to arrive at a solution comparing favourably with experiment. The approximately true equation of the impulse valve motion is

$$\frac{d^2s}{dt^2}=\frac{2Hg^2t}{L}\left[1-\frac{W\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}{2AHw\Phi(s)}\right]\Bigg/\sqrt{\frac{Wg}{Aw\Phi(s)}} \quad . \quad (54)$$

which gives finally

$$t_3 = \sqrt[3]{Ls\sqrt{\frac{Wg}{Aw\Phi(s)}}\Bigg/ 4g^2H\left[1-\frac{W\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}{2AHw\Phi(s)}\right]} \quad . \quad . \quad . \quad (55)$$

The variation of velocity of water during period 3 is too obscure to be expressed mathematically. Because of the short duration of the time, t_3, it is certain that this velocity does not vary greatly from its initial value, v_2, and for all practical purposes it can be assumed that $v_2 = v_3$. On this basis, the quantity of water flowing to waste during period 3 can be estimated as

$$Q_3 = \frac{w\pi D^2 v_2 t_3}{4} =$$

$$\frac{w\pi D^2}{4}\sqrt[3]{Ls\left(\frac{Wg}{Aw\Phi(s)}\right)^2\Bigg/ 4g^2H\left[1-\frac{W\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}{2AHw\Phi(s)}\right]} \quad . \quad . \quad . \quad (56)$$

The kinetic energy of the column of water arrested at the end of period 3 can be determined as

$$\frac{1}{2}\,\frac{w\pi D^2Lv_3{}^2}{4g} = \frac{w\pi D^2Lv_2{}^2}{8g} = \frac{\pi D^2LW}{8A\Phi(s)} \quad . \quad (57)$$

Analysis of Period 4. At the end of period 3 the impulse valve is completely closed, and abrupt retardation takes place causing the delivery valve to open, whilst at the same time the velocity in the drive pipe is reduced in proportion to the pressure head $(h+h_r)$ generated in excess of the static head H. The duration of period 4 is very short and may be estimated to be

$$t_4 = 2L/C \quad . \quad . \quad . \quad . \quad . \quad . \quad (58)$$

That is, the time required for a complete reflection of the acoustic wave travelling along the drive pipe. The velocity of the acoustic wave in an elastic medium is given by the formula

$$C = \sqrt{144K_1g/w} \quad . \quad . \quad . \quad . \quad . \quad (59)$$

where K_1 can be determined from equation (34) or (35) due to Joukowski and Gibson respectively. Calculations prove that for ferrous materials the differences between equations (34) and (35) are too small to be significant, and it is proposed to use Joukowski's formula for K_1 as being the simpler of the two. Consequently, the duration of period 4 may be written as

$$t_4 = 2L\sqrt{\left(\frac{1}{K}+\frac{D}{Et_p}\right)\frac{w}{144g}} \quad . \quad . \quad . \quad . \quad (60)$$

If the velocity of the water at the end of period 4 is v_4, then there must be a relation

$$v_4 = v_3-(h+h_r)g/C \quad . \quad . \quad . \quad . \quad (61)$$

From the above equation the maximum pressure head which can be developed in an hydraulic ram can be determined. Obviously, when $v_4 = 0$, $h_r = 0$ and

$$h = h_{\max} = \frac{Cv_3}{g} = \frac{C}{g}\sqrt{\frac{Wg}{Aw\Phi(s)}} = \sqrt{\frac{144W}{\left(\frac{1}{K}+\frac{D}{Et_p}\right)Aw^2\Phi(s)}} \quad (62)$$

Equation (62) shows that for a given weight of the impulse valve W and length of stroke s, there is always a certain maximum delivery head beyond which no water can be pumped. It also shows that if the impulse valve is too light, only small delivery heads can be obtained.

In consequence of the reduction of velocity of water during period 4, there is a certain loss of kinetic energy which is expended in compressing the water contained in the drive pipe and stretching the walls of the latter. This loss of kinetic energy can be determined as

$$\frac{w^2(h+h_r)^2\pi D^2L}{8\times 144}\left(\frac{1}{K}+\frac{D}{Et_p}\right) \quad . \quad . \quad . \quad (63)$$

Analysis of Period 5. The delivery valve is now fully open, and the energy available for pumping the quantity of water q against the head $(h+h_r)$ equals the kinetic energy developed at the end of period 3 (equation (57)) less the energy of strain expended during period 4 (expression (63)). The quantity of water delivered per cycle is therefore

$$q = \frac{\pi D^2L}{8(h+h_r)}\left[\frac{W}{A\Phi(s)}-\frac{w^2}{144}\left(\frac{1}{K}+\frac{D}{Et_p}\right)(h+h_r)^2\right] \quad . \quad . \quad (64)$$

Since, for any hydraulic ram installation the factors D, L, K, w, A, E, t_p, and $(h+h_r)$ are fixed, the result obtained in equation (64) shows that the quantity of water delivered depends principally on the weight of the impulse valve W and the factor $\Phi(s)$. This important fact was confirmed by experiments, which showed that for a given $\Phi(s)$, the increase of the valve weight caused the quantity q to increase.

The duration of period 5 can be determined from the principle of momentum, mass × change of velocity = impulse. On this basis it was found that

$$t_5 = \frac{L}{g(h+h_r)}\left[\sqrt{\frac{Wg}{Aw\Phi(s)}}-(h+h_r)\sqrt{\left(\frac{1}{K}+\frac{D}{Et_p}\right)\frac{gw}{144}}\right] \quad . \quad (65)$$

The result obtained shows that the time of delivery rapidly decreases with the increase of delivery head.

Analysis of Period 6. At the end of period 5, the whole of the energy available for pumping is exhausted, and the delivery valve closes. This closure of the delivery valve means that the drive pipe, which so far had remained under the total pressure head $(H+h+h_r)$, suddenly returns to the static head H. The time during which this change takes place is very short and can be estimated as a time required for a complete reflection of an acoustic wave, C, or

$$t_6 = \frac{2L}{C} = 2L\sqrt{\left(\frac{1}{K}+\frac{D}{Et_p}\right)\frac{w}{144g}} \quad . \quad . \quad (66)$$

As a result of the sudden change of pressure, the energy of strain of the water and drive pipe (expression (63)) is converted into the kinetic energy of the water moving towards the supply tank. The phenomenon is similar to that of rebound of elastic bodies, and is illustrated by Fig. 25. The photograph was taken after the level of water in the supply tank was reduced to a minimum and a pressure of approximately 150 lb. per sq. in. had developed in the experimental ram. The column of spouting water reaches the height of about 6 inches above the level of the static water, and some particles of water can be seen in the air.

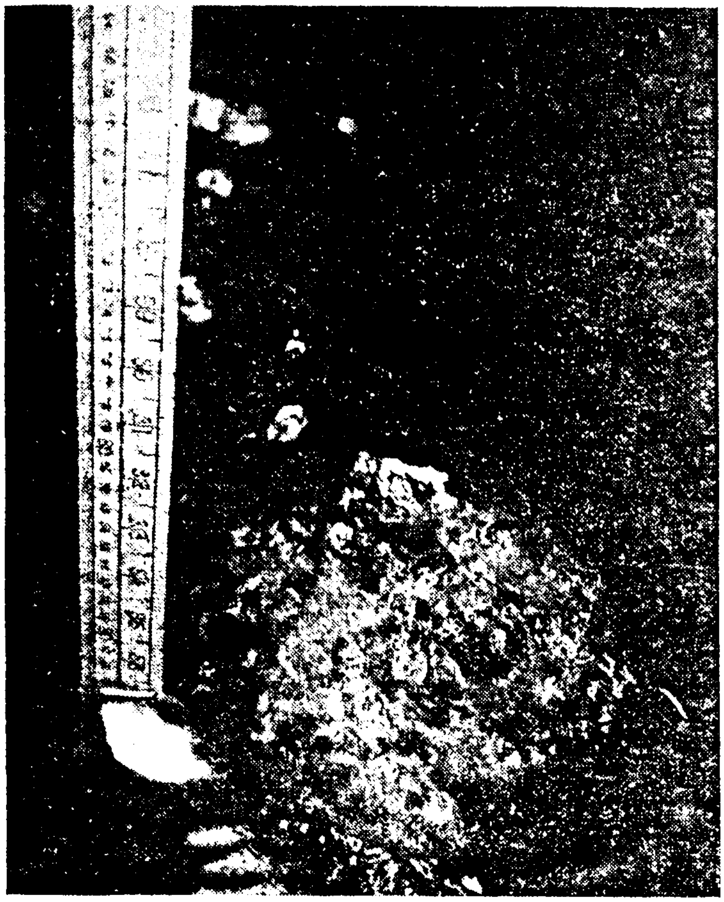


Fig. 25. Recoil Due to Water Hammer

The magnitude of the negative velocity with which the water recoils from the impulse valve can be obtained by equating the energy of strain to the kinetic energy:—

$$\frac{w^2\pi D^2L}{8\times 144}\left(\frac{1}{K}+\frac{D}{Et_p}\right)(h+h_r)^2 = \frac{w\pi D^2L}{8g}v_6{}^2$$

from which

$$v_6 = -(h+h_r)\sqrt{\left(\frac{1}{K}+\frac{D}{Et_p}\right)\frac{gw}{144}} \quad . \; . \quad (67)$$

Analysis of Period 7. At the end of period 6 the water in the drive pipe has acquired a negative velocity, consequently, the impulse valve opens under its own weight and a pressure difference caused by the recoiling water. The time t_7 during which the water moves away from the impulse valve may be found from the equation of motion, equation (42), given in the analysis of period 1, where losses of head are neglected on account of the shortness of period 7. The solution of the equation gives

$$t_7 = \frac{L(h+h_r)}{H}\sqrt{\left(\frac{1}{K}+\frac{D}{Et_p}\right)\frac{w}{144g}} \quad . \; . \quad (68)$$

The magnitude of the distance y travelled by the recoiling water inside the drive pipe can be obtained by equating the kinetic energy of the column of water to its potential energy. This gives

$$y = wL\left(\frac{1}{K}+\frac{D}{Et_p}\right)(h+h_r)^2/288H \quad . \; . \quad (69)$$

The reverse motion of water during period 7 subsequently affects the duration of period 1 and the quantity of water escaping during period 2 as indicated in equations (43) and (48) respectively.

SUMMARY OF THEORY DEVELOPED

The results of the foregoing analysis can be presented in five general formulae.

(I) The quantity of water delivered is

$$q = \frac{\pi D^2L}{8(h+h_r)}\left[\frac{W}{A\Phi(s)}-\frac{w^2}{144}\left(\frac{1}{K}+\frac{D}{Et_p}\right)(h+h_r)^2\right] \text{ pounds per cycle} \quad . \; . \; . \quad (70)$$

This formula shows that for a given W and $\Phi(s)$ there is always a maximum delivery head for which q becomes zero.

(II) The quantity of water flowing to waste is

$Q = Q_2+Q_3$ or

$$Q = \frac{w\pi D^2}{4}\left[\frac{L}{\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}\times \log_e\left(\frac{\dfrac{2gH}{1+\Sigma j+4f\frac{L}{D}+R(s)}-\dfrac{gw}{144}\left(\dfrac{1}{K}+\dfrac{D}{Et_p}\right)(h+h_r)^2}{\dfrac{2gH}{1+\Sigma j+4f\frac{L}{D}+R(s)}-\dfrac{Wg}{Aw\Phi(s)}}\right)+\sqrt[3]{\frac{sL\left(\dfrac{Wg}{Aw\Phi(s)}\right)^2}{4g^2H\left[1-\dfrac{W\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}{2AHw\Phi(s)}\right]}}\right]$$

pounds per cycle . . . (71)

An important feature of formula (71) is that it shows the interdependence of the factors H, W, $\Phi(s)$, $R(s)$, L, D, f, and Σj, since if

$$2gH\left\{1+\Sigma j+4f\frac{L}{D}+R(s)\right\} = Wg/Aw\Phi(s)$$

the impulse valve will not close at all, and Q becomes infinity.

(III) The duration of a total cycle is

$$T = t_1+t_2+t_3+t_4+t_5+t_6+t_7$$

For t_1+t_2 it is simpler to use equation (53). Adding and rearranging the terms already found, the value of T is

$$T = \frac{L}{\sqrt{2gH\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}}\times \log_e\left[\frac{1+\sqrt{\dfrac{W\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}{2AHw\Phi(s)}}}{1-\sqrt{\dfrac{W\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}{2AHw\Phi(s)}}}\right]+\sqrt[3]{\frac{sL\sqrt{\dfrac{Wg}{Aw\Phi(s)}}}{4g^2H\left[1-\dfrac{W\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}{2AHw\Phi(s)}\right]}}+\frac{L}{g(h+h_r)}\sqrt{\frac{Wg}{Aw\Phi(s)}}+L\left(\frac{h+h_r}{H}+3\right)\sqrt{\left(\frac{1}{K}+\frac{D}{Et_p}\right)\frac{w}{144g}}$$

seconds per cycle . . . (72)

(IV) According to the formula proposed by Eytelwein (1805) and Rankine (1872), the efficiency of the hydraulic ram is $\eta = \frac{qh}{QH}$

$$\text{or } \eta = hL\left[\frac{W}{A\Phi(s)}-\frac{w^2}{144}\left(\frac{1}{K}+\frac{D}{Et_p}\right)(h+h_r)^2\right]\Big/$$

$$2wH(h+h_r)\left[\frac{L}{1+\Sigma j+4f\frac{L}{D}+R(s)}\times \log_e\left(\frac{\dfrac{2gH}{1+\Sigma j+4f\frac{L}{D}+R(s)}-\dfrac{gw}{144}\left(\dfrac{1}{K}+\dfrac{D}{Et_p}\right)(h+h_r)^2}{\dfrac{2gH}{1+\Sigma j+4f\frac{L}{D}+R(s)}-\dfrac{Wg}{Aw\Phi(s)}}\right)+\sqrt[3]{\frac{sL\left(\dfrac{Wg}{Aw\Phi(s)}\right)^2}{4g^2H\left[1-\dfrac{W\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}{2AHw\Phi(s)}\right]}}\right] \quad . \; . \quad (73)$$

The above formula for efficiency is dimensionless and shows the effect of all the factors controlling the performance of the machine.

(V) The value of the power of the hydraulic ram (P_d) is $\frac{qh}{T}$, or

$$P_d = \pi D^2Lh\left[\frac{W}{A\Phi(s)}-\frac{w^2}{144}\left(\frac{1}{K}+\frac{D}{Et_p}\right)(h+h_r)^2\right]\Big/$$

$$8(h+h_r)\left[\frac{L}{\sqrt{2gH\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}}\times \log_e\left(\frac{1+\sqrt{\dfrac{W\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}{2AHw\Phi(s)}}}{1-\sqrt{\dfrac{W\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}{2AHw\Phi(s)}}}\right)+\sqrt[3]{\frac{sL\sqrt{\dfrac{Wg}{Aw\Phi(s)}}}{4g^2H\left[1-\dfrac{W\left[1+\Sigma j+4f\frac{L}{D}+R(s)\right]}{2AHw\Phi(s)}\right]}}\right.$$

$$\left.+\frac{L}{g(h+h_r)}\sqrt{\frac{Wg}{Aw\Phi(s)}}+L\left(\frac{h+h_r}{H}+3\right)\times \sqrt{\left(\frac{1}{K}+\frac{D}{Et_p}\right)\frac{w}{144g}}\right] \text{ ft.-lb. per sec.} \quad . \; . \; . \; . \; . \quad (74)$$

The formula for power is not less important than the formula for efficiency, which does not take into account the rate of pumping.

EXPERIMENTAL VERIFICATION OF THEORY DEVELOPED

To verify the theory developed, some 500 experiments were made on the experimental apparatus described above. The majority of the experiments were devoted to the investigation of the effect of the length of the impulse valve stroke and its weight. With various combinations of these two factors, it was possible to vary the characteristics of the machine over a very wide range of delivery heads. Experimental results were compared with the theoretical computations obtained by the employment of the final formulae (70)–(74). In these calculations, the following factors were kept constant throughout:—

A	0·043 sq. ft.	K	300,000 lb. per sq. in.
D	0·166 feet.	L	60·75 feet.
E	28,000,000 lb. per sq. in.	t_p	0·031 feet.
g	32·2 ft. per sec. per sec.	w	62·4 lb. per cu. ft.
H	13·0 feet.		

$$1+\Sigma j+4f\frac{L}{D} = 27{\cdot}1.$$

The values of $R(s)$ and $\Phi(s)$ used in the above calculations are shown in Fig. 26. The head lost during the period of retardation was calculated from equation (40) in which $h_{\max}$ was determined by means of equation (62). The comparison of theoretical and experimental data indicates that the agreement is very close over a wide range of operating conditions, and that the maximum difference between the theory and experiment does not in the least favourable circumstances exceed 15 per cent. This is shown in Fig. 26*a*, *b*, and *c*, which show graphically typical theoretical and experimental data as obtained for the three basic elements of performance of the machine.

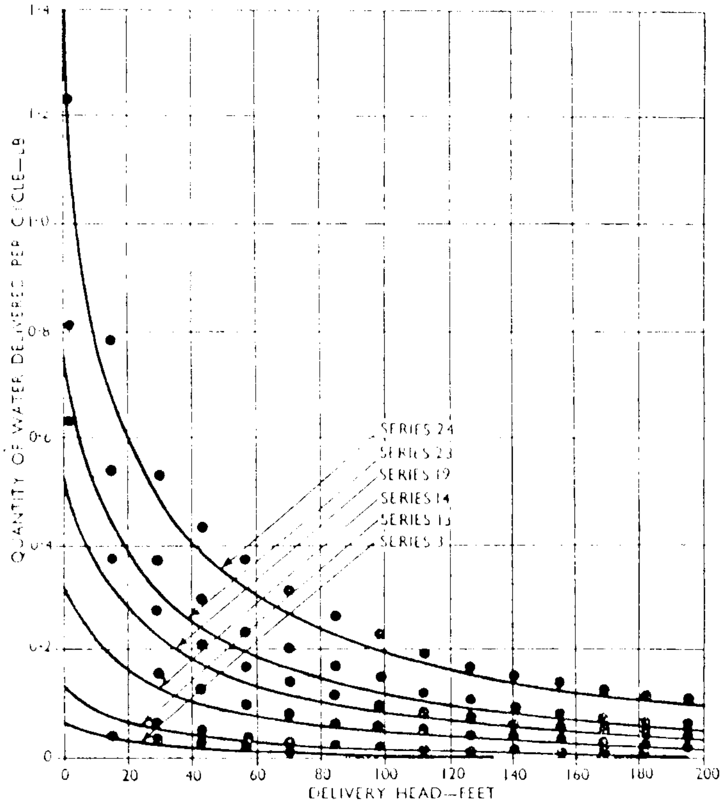
Considering that in the theory developed assumptions were made that: (1) the factor $\left(1+\Sigma j+4f\frac{L}{D}\right)$ remains constant; (2) there is no friction during periods 1 and 7; (3) velocity v_2 equals velocity v_3; (4) during the recoil, the water behaves like a solid bar; and (5) there is no effect of an air chamber on the quantity of water delivered—it must be concluded that the cumulative deviation between the theory and experiment gives very satisfactory correlation. Without the above assumptions, the theory developed would have been considerably more complicated, and the final formulae (70)–(74) would have been too involved to be of any practical use.

Experiments with the larger air chamber proved that the size of an air chamber affects the quantity of water delivered. This can be explained in the following manner. If q is the quantity of water delivered during a single cycle, the duration of which is T, then the average rate of flow through the delivery pipe is q/T. However, the time of actual pumping is much shorter than the total time T, and the air chamber receives water at a much quicker rate q/t_5. Since the water is delivered at the average rate q/T, the excess of water temporarily stored in the air chamber during the time t_5, is $q_c = q(1-t_5/T)$. It is now clear that in addition to the kinetic energy generated during the period of acceleration, there is an additional source of energy available for pumping, namely, the potential energy of the quantity of water q_c descending from the supply tank into the air chamber during period 5. If h_c is the level of water in the air chamber over the impulse valve, then this energy may be estimated as $q(1-t_5/T)(H-h_c)$. Compared with the kinetic energy developed in the drive pipe, the additional potential energy due to the effect of air chamber is small and to avoid unwarranted complications in the theory developed, it was ignored.

CONCLUSIONS

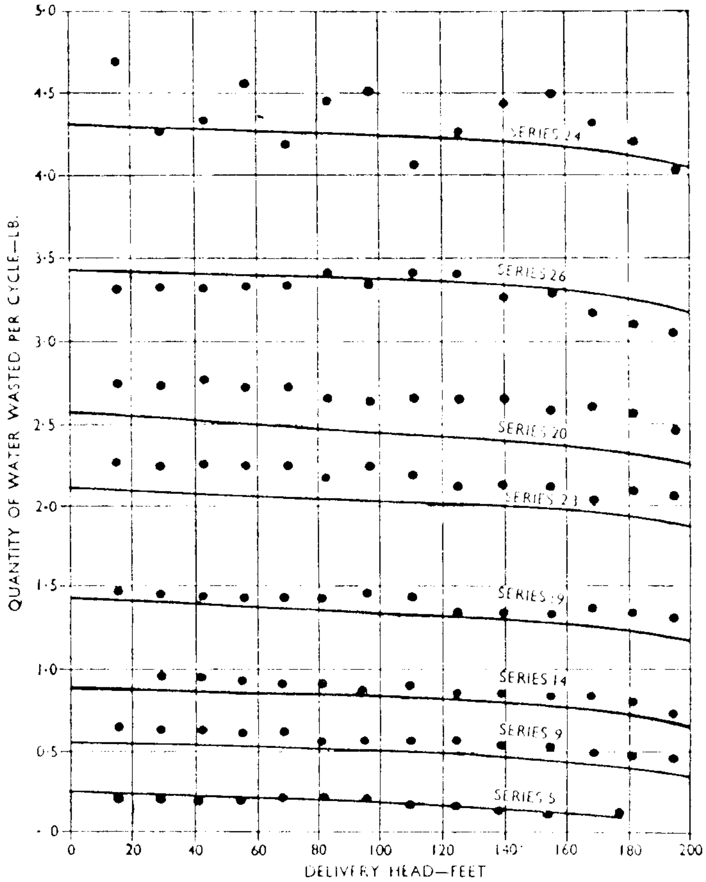
The results obtained in this investigation can be summarized in the following eight points concerning the theory developed.

(1) On the general laws of mechanics, it is possible to forecast the behaviour of any automatic hydraulic ram, provided that the following four properties of a given installation have been determined separately by experiment: (*a*) factor of loss of head in the drive pipe $\left(1+\Sigma j+4f\frac{L}{D}\right)$, (*b*) factor of loss of head due to the impulse valve, $R(s)$, (*c*) drag coefficient of the impulse valve, $\Phi(s)$, and (*d*) head lost during the period of retardation, h_r.



a Quantity of water delivered per cycle.

Series	24	23	19	14	13	3
Values of h in feet when $q = 0$	590·4	467·5	394·1	302·1	193·2	135·4



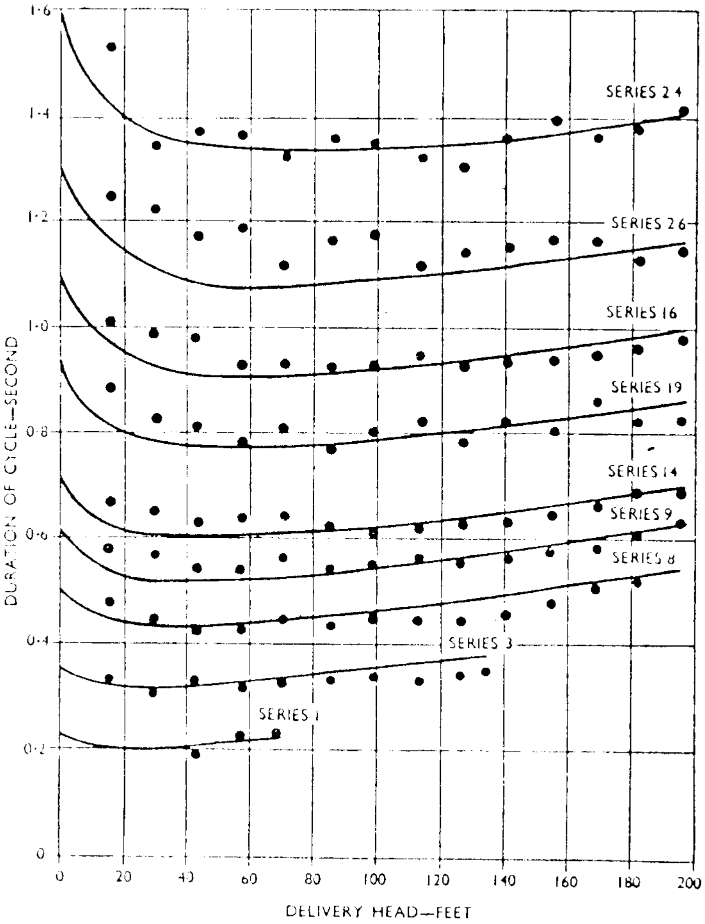
b Quantity of water wasted per cycle.

(2) Five formulae are developed expressing quantity of water delivered per cycle q, quantity of water wasted per cycle Q, duration of cycle T, efficiency η, and power developed P_d—as a function of numerous variables entering into the problem. If all the variables, except the delivery head, are fixed it is possible to express q, Q, T, η, and P_d as a function of the former. The resulting set of five curves (Fig. 27) constitute the characteristics of an hydraulic ram.

(3) That efficiency is not the only criterion required in the estimate of the performance of the hydraulic ram is stressed. The hydraulic ram may be so adjusted that it will work with reasonable efficiency and yet the rate of pumping, or, what is analogous—its power, may be negligible.

(4) The utmost importance of the proper selection of the impulse valve load is stressed. As shown in Fig. 28*a* and *b*, an increase in the impulse valve load consistently increases the power of the machine and its capacity to work under the increased delivery head. Further, an increased load increases the efficiency of the machine up to a certain limit, after which it decreases, tending to zero. The effect of the impulse valve load is closely connected with the length of the impulse valve stroke. In general, an increase in the length of the impulse valve stroke decreases the permissible load on the impulse valve. This is due to the marked decrease of the coefficient of drag.

(5) A solution of the problem, hitherto unsolved, of the desirable length of the drive pipe is provided. Fig. 29 shows the effect of the length of the drive pipe on the performance of the machine, determined for given conditions. It can be seen, that for other factors remaining constant, an increase in the length of the drive pipe: (*a*) causes proportional increase in the quantity of water delivered per cycle, (*b*) increases the quantity of water wasted per cycle, (*c*) increases the duration of the cycle, and (*d*) causes efficiency and power to increase, pass through a maximum, and subsequently decrease to zero.



c Duration of cycle.

Fig. 26. Theoretical and Experimental Relations Between Delivery Head and the Three Basic Elements of Performance

———— Theoretical. ● Experimental.

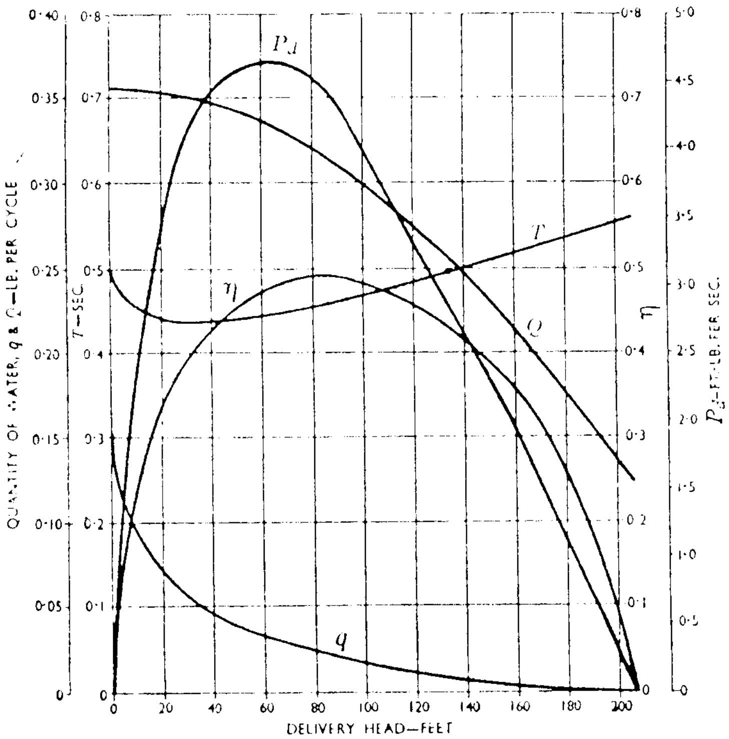


Fig. 27. Typical Characteristics of Hydraulic Ram

For conditions of series 8.

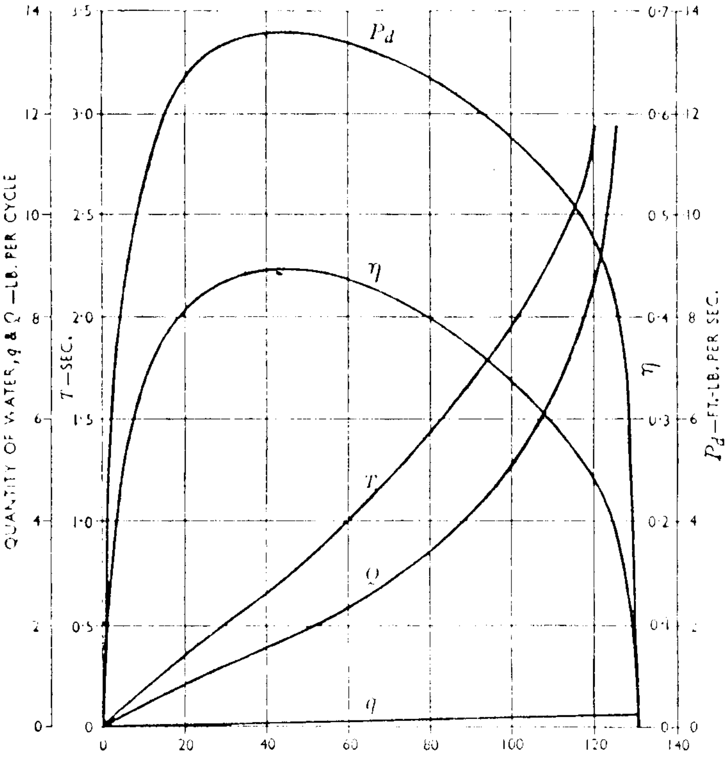
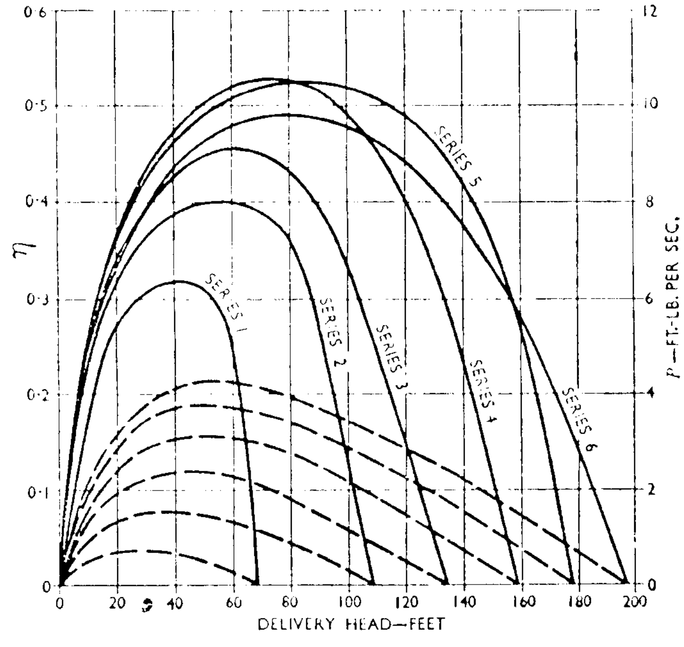


Fig. 29. Effect of Length of Drive Pipe on Performance of Hydraulic Ram

Series 20, h = 112·6 feet.

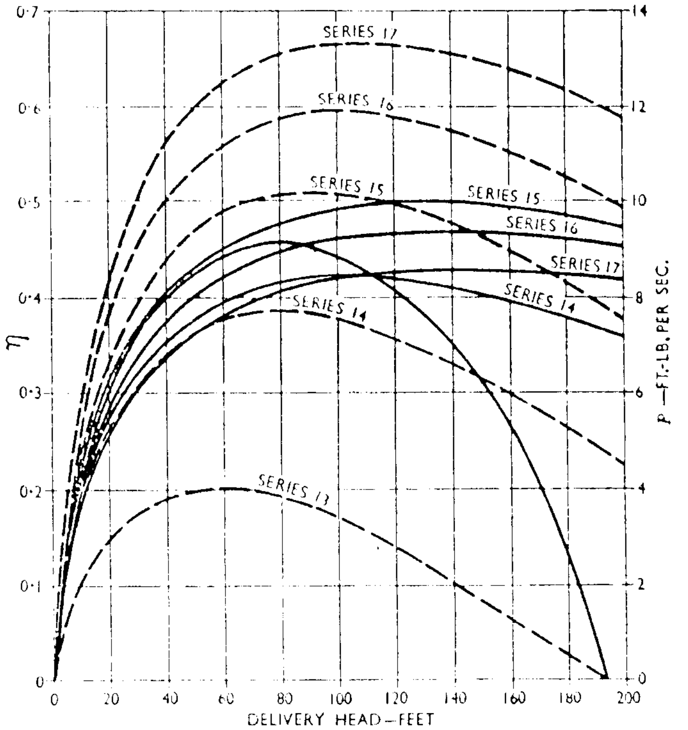
(6) A basis for a rational design of the hydraulic ram installation is established. Once the four experimental factors required by the theory are known, numerous characteristics of the proposed installation can be computed and consequently selection of the most advantageous set of conditions can be made.

(7) It was ascertained experimentally that the variation of the diameter and grade of hardness of rubber disks used in the delivery valve had a somewhat minor effect on the performance



a $s = \frac{1}{16}$ inch.

Series .	1	2	3	4	5	6
W, lb. .	0·605	1·485	2·365	3·245	4·125	5·005



b $s = \frac{3}{16}$ inch.

Series . . .	13	14	15	16	17
W, lb. . . .	0·605	1·485	2·365	3·245	4·125

Fig. 28. Effect of Impulse Valve Load on Efficiency and Power

——— Efficiency. - - - - - Power.

of the hydraulic ram. These experiments seem to indicate that the diameter of the rubber disk covering the ports of the delivery valve should not be excessive, and that for increased delivery heads, an increased hardness of rubber is desirable.

(8) Experiments with the larger air chamber resulted in a consistent increase of approximately 10 per cent in the efficiency and power developed. As explained, this is due to the additional potential energy available for pumping formed by water temporarily stored in the air chamber during the period of retardation.

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APPENDIX

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Discussion

Mr. G. A. WAUCHOPE, M.I.Mech.E., who opened the discussion said in connexion with the paper by Mr. Binnie and Dr. Thackrah that water hammer in a pumping main and its prevention were of great importance and, while there had been a good deal of discussion about it in the United States, not much had been done in Great Britain.

In practice, it was not absolutely essential to be able to accurately calculate the pressure surges. It was essential to know, however, what type of protective measures must be taken and when they must be taken, to keep the pressure surges below the maximum safe working pressure of the pipes and fittings.

Fig. 30 showed the problem which confronted the designers of pipe-lines and pumps. He had shown a curve without a scale suggesting the relation of the pressure surge to the head and length of pipe. He had divided the problem into three types: type 1 was one extreme, in which the static head—the distance through which the water had to be raised from the suction level to the discharge level—was great in proportion to the length of the pipe. At the other extreme was the reverse condition, type 3, where the distance between the suction level and the discharge level was relatively small but the pipe might be of considerable length. When the pipe-line was short compared with the static head (type 1) the problem was due to the quick reversal of flow. A non-return valve must close before the return flow commenced. That was not always possible and, consequently, the non-return valve flaps might be slammed with great violence and much damage done.

Under type 2 conditions the state of affairs was easier, there the amount of kinetic energy in the pipe was such that the flow continued gently, and the water drawn through the pump was not at a high velocity. Partition of the water column did not take place, the return flow did not slam the reflux valve, and the reflection of pressure occurred many times before the valve closed. That was probably the most common arrangement, and might explain why the problem had not always received the necessary attention.

The authors' type of slam in the electronic apparatus had shown that the rise in pressure might be high if no air were admitted into the pipe. He suggested that the effect of letting air into the pipe was to produce a curve of the type shown dotted on the right of Fig. 30; after a certain increase in length, for a given head the admission of air began to reduce the pressure surge until, when the pipe was long enough with respect to the head, the air might almost entirely remove the pressure surge. He doubted whether the position at which the dotted curve went to zero was correct, but for practical purposes under those favourable conditions the pressure surge could probably be neglected.

He asked whether the authors could indicate the position of point A on Fig. 30, and whether they could insert a scale. That might be too difficult, but it was a matter of importance to those responsible for the design of pumping equipment and pipe-lines where electric motors, which stopped suddenly, were involved. The shape of the curve could be correct for a certain velocity condition. If, however, for type 1 conditions the velocity were higher, the flow of water would cease less rapidly, and possibly the pressure surge would be less, whereas, for type 3 in the higher velocity condition the surge was reduced when air was drawn in.

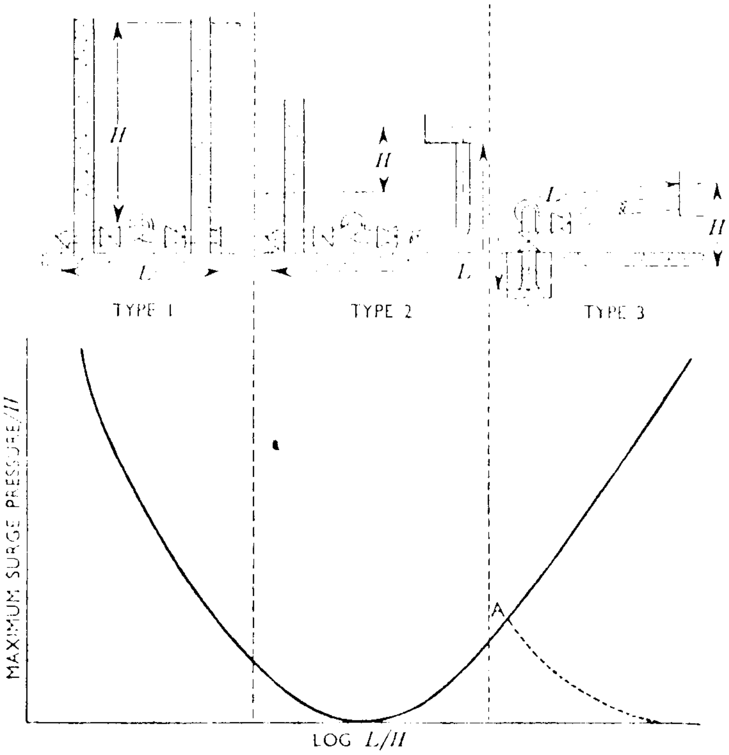

Fig. 30. Varying Arrangement of Pipe-lines and Pumps

Type 1, *H* large in proportion to *L*.
Type 2, intermediate condition.
Type 3, *H* small in proportion to *L*.

Also, under type 3 conditions the pumps must take some definite time to come to rest. He wondered whether the authors agreed that would have the effect of increasing the pressure surge, because the average change of velocity during the stopping period was reduced, so that the quantity of air drawn in was smaller and therefore its cushioning effect would be less.

An interesting application of the admission of air to a pipe-line was in connexion with suction dredging, where spoil was drawn up by means of a suction pipe, and discharged through considerable lengths of piping. The operator might allow the suction pipe to get too near the ground, and be sucked down on to the ground when immediate shutting off would be nearly attained. The danger of pressure surge was overcome by putting a non-return valve on a T-piece in the discharge pipe-line near the pump, so that it was kept closed when the pump was operating properly and opened as soon as the pressure in the pipe-line fell below that of atmospheric pressure, thus admitting air to the pipe. If that were not done, the casing of the pump might be split.

Fig. 31 showed the contour of a pipe-line which represented a familiar type of pumping main. Above the contour was the hydraulic gradient when the pump was working. Air valves were fitted at the highest points to prevent air locks. He asked the authors' views on what relief might be obtained in such an arrangement by the air being permitted to enter the pipe at the normal air release-valve positions.

Various graphical methods of dealing with pressure surges had been published which had the defect in most cases of assuming that lower pressures than could actually be achieved occurred at the first reflection. He asked the authors' views on the application of those methods where air could be admitted.

An attempt could be made to overcome or minimize surges by employment of a by-pass to the pump, incorporating some form of non-return valve (authors' method (b)). It had the effect of permitting water—instead of air, as in the authors' experiments—freely to enter the pipe, and he would be interested to have comments on the advantages of that method. Many engineers did not advocate admission of air to water mains because of possible corrosion. He wondered whether it was considered reasonable to assume that the disadvantage might be minimized, if not eliminated, if the pipes were made of steel or cast-iron lined with concrete or asphalt.

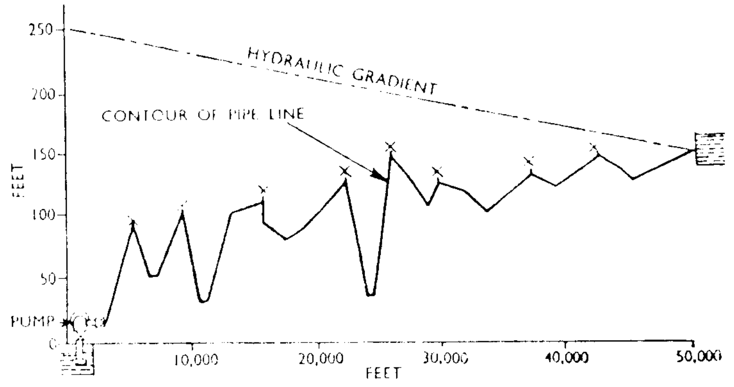

Fig. 31. Pipe-line Contour

× Air-valve positions.

He thought that Dr. Krol's paper might lead to other developments. The automatic hydraulic ram was a device which had been in use for many years, but had not been very generally applied.

Consideration might be given to increasing the frequency and applying the principle to produce small flows of liquid at high pressures from sources of low head and high quantity; pump designers were always trying to achieve that in the most economic way.

Mr. H. R. LUPTON, M.C., M.A., M.I.Mech.E., said, in regard to the paper by Mr. Binnie and Dr. Thackrah, that the method of preventing water hammer in pumping mains which the authors had developed particularly was the admission of air into the main. Air in long pumping mains was an anathema to water engineers; they knew that when a main had not been properly de-aerated it would probably burst.

There were often accumulations of air at the high spots in the main, and one of those accumulations might move so that there was a sudden change in the velocity of the water. The moving mass of water might come up against another bubble banked up and, being rapidly checked, cause a high pressure. It was known that air admitted into mains was liable to result in high pressures in unknown and unpredictable ways, so that he would rather not adopt that method of preventing water hammer.

The last of the six methods of preventing water hammer given by the authors was to put a by-pass round the pump and its valves, which should allow water to follow the retreating column in the long delivery main and so prevent cavitation if the driving agent stopped. He would prefer to say prevent "separation", because there might be a definite separation of the long water column with no water to follow behind it. The by-pass method could be applied only where the supply was within suction distance of the main; if a pump in a well stopped, probably the atmospheric pressure would be insufficient to cause the water to follow through the by-pass and prevent it separating from the retreating column.

At Ashford Common the Metropolitan Water Board had 5,000 h.p. of grid-driven pumps. Delivery was effected through several large mains. To reduce the probability of bursting through "separation" considerable trouble to provide safeguards was worth while. No provision was made to prevent ordinary surges.

Each of the pumps at Ashford Common delivered through a hydraulically operated valve. The valves would be shut when the pumps started and, normally, shut before the pumps stopped; therefore, non-return valves were not necessary. They took the place of both sluice and non-return valves. They shut and opened at a controlled slow speed under the action of hydraulic power, independent of electric power. That would normally cause the pumps to stop and start without introducing any undue pressure or surge on the main. In the event of the electric power failing without warning, the valves might shut down more quickly than the column of water ceased to flow, but a by-pass must be provided in any case, because the resistance imposed by a pump against continued forward flow, when the driving agent stopped, was such that separation would occur. Atmospheric pressure would not continue to impel water through an undriven pump at the rate at which the water was retreating along the main.

With those particular pumps, at any rate in the two largest sizes, the time taken between the failure of electric power and the differential pressure across the pump falling to zero was about 3 seconds. If the authors' method (4) were adopted, and flywheels were put in, the period could be increased to more than 3 seconds, but the Board had not been recommended to do that.

A by-pass valve was installed in parallel with each group of pumps, one for every delivery main, placed in a by-pass round the three pumps (there were three pumps in a group) and set to open in a period of about 2 seconds under hydraulic action in the event of a failure of electric power, and to remain open for a period of about 70 seconds in the case of the longest of the mains delivering to the farthest destination; that being the period during which the forward flow went on under the action of momentum, opposed by gravity. The valve would then begin to shut gradually, so that there would be no excessively high pressure. The period during which the valve could shut was adjustable, but was long. There was usually no difficulty in disposing of the water, which in an emergency would have to be let in backwards into the suction well.

It might be objected that hydraulic valves might not work, and that some device was needed in the station which was more reliable and safer than a valve operated by external means, even hydraulic. He also held that opinion. To overcome that another valve which did not depend on external hydraulic agency for operation had been put in parallel with each of the by-passes to each group of pumps. He called it an "unbalanced opening, balanced closing" valve. It was merely an ordinary flap valve placed vertically, the two flaps of which, when they were opened by the forward flow of the water, would push up a sleeve by means of two rods and form an alternative path for the water. When the forward velocity through the valve ceased, the flaps would fall shut again, but there was lost motion between them and the sleeve, so that the sleeve need not fall shut at the same time, and, in fact, would not do so, because it was held up by a dashpot. The dashpot was set to allow the sleeve valve to shut only at such speed as would never impose excessive pressure on the main.

Mr. P. LINTON, B.Sc., A.M.I.Mech.E., congratulated Mr. Binnie and Dr. Thackrah on their clear demonstration of the advantages of injecting air into a pipe-line when the flow was suddenly stopped. The surge pressures had been reduced to approximately one-tenth compared with the unprotected pipe, and the inadequacy of a relief valve in reducing the surges was clearly apparent.

The authors had shown considerable ingenuity in obtaining mathematical expressions for the main design values, but the complexity of those expressions would prevent many engineers from giving that method of surge protection the favourable consideration it deserved. He referred in particular to equation (2) which gave the volume of air absorbed, and equation (3) which gave the time taken for the column to come to rest; equation (6) was an implicit relation from which the maximum pressure might be obtained. Equations (2) and (3) were not easily solved when dealing with a large number of cases. The authors had mentioned that the solution of equation (6) would be greatly facilitated by the tables published by the Federal Works Agency; that would probably not make it popular with water engineers and others interested in the solution of the problem.

He would like, therefore, to direct attention to a method by which a solution could be obtained with only slide-rule and set-square. It was the Schnyder-Bergeron graphical method, which made use of a head–velocity diagram in which the conditions at the two ends of the pipe-line were plotted as the boundary curves. One of the boundary conditions was the combined static and friction characteristic, which was equivalent to the curve for pump head shown in Fig. 5.

In Fig. 32 the initial condition might be represented by the point A. When the flow stopped suddenly, the pressure and flow fell in such a manner that they were represented by points on a straight line through A of slope $\Delta h/\Delta v = a/g$, a being the velocity of the surge waves in the pipe. The head would drop until the setting of the air valve was reached. The condition at that end, the second boundary condition, was given by the horizontal line corresponding to the head represented by the setting of the air valve. The column would travel on, and its velocity would slowly be diminished by means of surges passing backward and forward; those were represented in Fig. 32 by further straight lines, surge lines, which went backward and forward between the two boundary curves in the manner shown; the slope, alternately positive and negative, being given again by a/g.

It was then possible to evaluate directly the time taken for the column to come to rest and the volume of air drawn in. The time taken in the example chosen—6 ft. per sec. initial velocity and 7·3 feet of water valve loading—was 18 pipe periods or 1·33 seconds, compared with the 1·29 seconds calculated by the authors. The distance travelled by the column, which could be obtained by multiplying the time of each period by the velocity during that interval, gave 0·0691 cu. ft. of free air, compared with the figure of 0·0624 cu. ft. obtained by the authors.

When the water column reversed, the air valve closed and the air contained in the pipe would act as an air vessel. A construction for that case had been given by the French engineer Bergeron (1950)*. While the surge lines were continued as before, so-called fictitious surge lines were drawn for the "transit of the surge through the air space to the closed valve", the boundary condition of which was represented by the axis of heads (zero flow). The slope of the imaginary surge lines was given by $\Delta h/\Delta v = AtH^2/2V_0H_0$, where A was the cross-sectional area of the pipe, t the time interval for which the construction was carried out (in that case, one pipe period), H the absolute head during the period, and V_0 and H_0 the initial conditions, volume and head, of the air in the air space. A point representing the pressure and velocity at the air end of the water column was thus obtained from the one a period previous by drawing a true surge line which was reflected from the discharge end of the pipe, and a fictitious surge line reflected from the closed valve.

The only variable in the expression for the slope of the fictitious surge line was H, which varied as the pressure rose, and it was sufficiently accurate with that method to change that in steps of, say, 5 feet of water without seriously affecting the accuracy of the result. The construction could be continued as long as desired; it would give pressure, velocity, position of the water column, and time for any part of the oscillation. In particular, the maximum surge pressure was found to be 78·2 feet of water absolute, which compared with 81·4 feet calculated by the authors, and an experimental value of 72·8 feet.

The graphical method involved the same assumptions as those on which the authors had based their calculations and it could not, therefore, produce more accurate results. In general, however, it was quicker, while being sufficiently accurate, and it provided a direct picture of the changes in flow taking place. It would also be possible without great difficulty to take account of other factors, such as slow closing of both gate valve and air inlet valve, the effect of pump characteristic, and other conditions which would make a mathematical solution almost impossible.

The authors' air valve was light and comparatively large, and it would be interesting to investigate how the standard type of air valve normally fitted to pipe-lines would react to such a breaking

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* BERGERON, L. 1950 "Du Coup de Bélier en Hydraulique au Coup de Fondre en Electricité", p. 58 (Dunod, Paris).

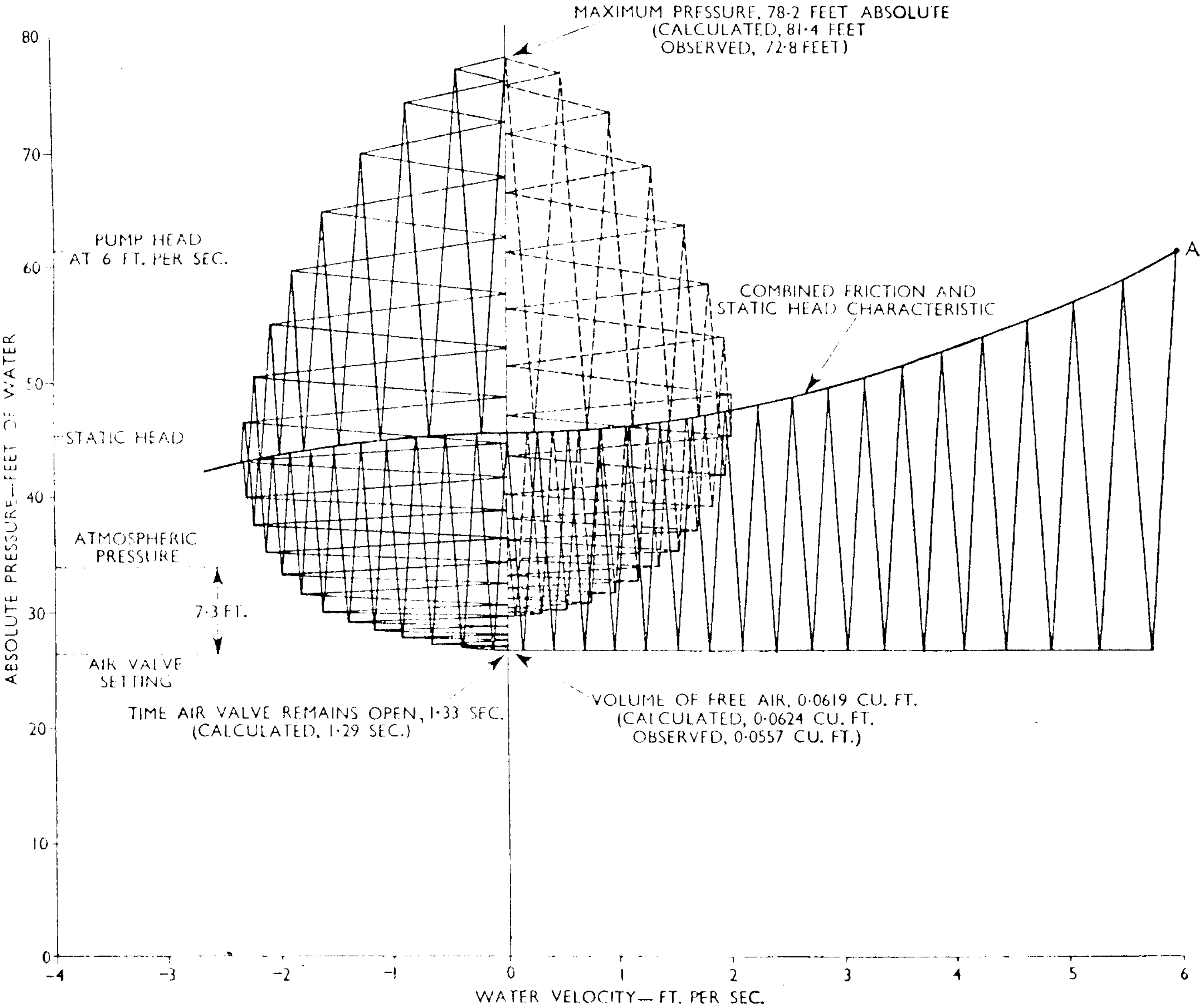

Fig. 32. Head–Velocity Diagram for Solution of Equation (6)

Period of pipe-line, 0·0739 second; a = 4,450 ft. per sec.

of the water column. The reflection of surge waves at a change of section, as expressed in formulae (20)–(22), was first described by Löwy (1928)*.

He would be interested to have views on the injection of air into a pipe-line. His experience had been that water engineers were anxious to avoid the presence of even small quantities of air in a main. To overcome that objection it might perhaps be possible to form the air cushion in a separate chamber connected to the pipe by a quick-acting valve.

Mr. H. R. Green (Winchester), commenting on Dr. Krol's paper, said that a 2-inch ram would give its highest efficiency when using from 10 gal. per min. down to 3 gal. per min. Simple types of ram would work with practical efficiency, using rubber valves, with a ratio of fall/lift varying from 1/2½ to 1/30 or 1/40. With compound rams with differential pistons any ratio might be obtained up to 1/80 to 1/100. Compound rams had not been considered. They employed the same system as the simple hydraulic ram, but instead of the blow forcing the water through directly to the delivery main it forced up a double-acting piston, one end of which was really a single-acting pump. Dirty water could be used to operate the ram, and clean water could be sucked up and delivered in the normal way.

His firm had made and installed over the years a number of rams to work with as small a supply as ½ gal. per min.; they were fitted with metal outlet valves with a rubber buffer to compensate for the low suction.

The author had stated that the use of those rams had not been as widespread as was warranted, but they were in use all over Great Britain, in India, and elsewhere. His firm had one which had been in existence for 126 years and was still giving the same service as when originally installed.

There were two types of compound ram; the double-acting piston type, and the type with a rubber tube in the centre and a gunmetal cage on the outside. The blow of the ram opened and closed the "breathing" tube, and the clean water was circulated on the outside. As the blow forced the rubber up against the cage the water was delivered, and on the recoil it was sucked in. That type could not be used where it was essential that the dirty water should not mix with the clean, because if the rubber tube were punctured contamination would result.

It had been found by experience that the weight of the impulse valve did not affect the efficiency, provided the ram would pass the desired quantity of water. It had been said that certain weights affected the efficiency of the ram, but in practice they had not found that to be so.

The correct length of drive pipes should always be based upon the ratio of the fall to the lift, i.e. the length for a fall of 10 feet forcing to an elevation of 100 feet should be approximately the same as for a fall of 30 feet forcing to an elevation of 300 feet. To decrease the loss by friction the length should be as short as possible consistent with ability to force to the given elevation. It should be on an even gradient throughout.

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* Löwy, 1928 "Druckschwankungen in Druckrohrleitungen" (Springer, Vienna).

Mr. M. O. W. Wolfe (Farnborough) said, with reference to the paper by Mr. Binnie and Dr. Thackrah, that he had been interested to find that with the actual measurement of the ram, the "top hat" diagram with corrugations on top, Fig. 14, showed no signs of "organ pipe" resonances in the passages of the pressure gauge, or resonances of a mechanical nature of the moving parts of the system. He had been surprised because the rate of change of pressure was very high indeed, as far as he could tell from the diagram. Possibly the absence of resonances in the pipe of the pick-up was due to the presence of the grease which the authors put into the tubes.

Mr. H. D. Manning, B.Sc. (London), said that previous speakers seemed to regard the subject of water hammer as connected purely with clean water, whereas the problem cropped up with sewage, where the conditions were slightly different. In most sewage pumping stations the quantities were small; rising mains were short; there was usually a substantial head, and Mr. Wauchope's second type of condition usually applied.

The site of a town with which his firm was concerned was very flat, and for various reasons large pumping units had been installed in sewage pumping stations and were fitted with small pump wells, resulting in frequent stopping and starting of the pumps. In a typical case the rising main was up to 5,000 yards long and 54 inches in diameter, with a static head of about 20 feet; there were up to 10 or 12 starts per hour.

The theoretical analysis on the basis suggested by the authors showed that, in at least three instances, there was risk of damage to mains and non-return valves by water hammer, and it was suggested that the admission of air might eliminate the risk. Flap valves, 6 inches in diameter, with an adjustable counterweight to get different conditions of pressure at which air was admitted, were provided in each case.

It might be assumed that those valves fulfilled their purpose, since the stations had been running for some months, and some for over a year, and he had not heard of any trouble. That evidence was negative, however, in so far as he had not personally verified that the valves were even installed. Each air-admission valve had been provided with an isolating sluice valve, and, although clear instructions were given that those valves should be left open it might be found that the valves had been shut all the time.

Mr. H. W. Puttick, M.I.Mech.E., said that the East Indian Railway had installed a rising main for pumping about 10,000 gal. per hour of water from a tube well in the United Provinces. There was an 8-inch main, 1½ miles in length and the static head was 150 feet. The vertical-spindle pump was protected by a non-return valve and a large air bottle inside the pump house. A power failure occurred while the pump was on its trial run, and the main was completely carried away at the right-angled bend outside the pump-house.

Four non-return valves were subsequently fitted in the pipe at intervals of 400–500 yards. There was still considerable water hammer—not dangerous water hammer, but heavy knocking. It was then decided to drill each non-return valve with a small hole to obtain visual indication of the depth of the water in the overhead tank from a special pressure gauge in the pump-house. The small holes completely eliminated the knocking in the rising main. Perhaps the authors could give an explanation for that.

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Mr. T. P. Wakeford (London) observed that water engineers tried to send only sterile water from the works. If they let foreign matter get into the mains, they could not be certain that the water was pure. The air in a pumping station might not be sterile, but it would probably not cause any contamination. The pressure in the main, however, might go below atmospheric pressure when it would be possible for contaminated water outside the main to get in through a leaky joint or a leak in the pipe. If, as had been suggested, automatic air valves were provided on the length of line of main and used to admit air, the pits in which they were situated might be full of contaminated water which could enter the main and cause trouble later.

Mr. T. E. Beacham, M.I.Mech.E., said that he would like to mention a problem in power transmission which was similar to the one dealt with by Mr. Binnie and Dr. Thackrah, in the hope that they could offer a solution. It concerned the pressure surge from a loaded hydraulic cylinder at the moment when it was open to exhaust. The main cylinder, at the instant before it was opened to exhaust, was full of liquid under pressure. The liquid was compressed by the pressure, hence a large amount of energy was stored in it. That energy, as well as the strain energy in the columns and cylinder walls, was released when the valve was opened, and might cause an extremely heavy hammer blow. One method of preventing the blow was first to open up the cylinder to a restrictor, but conditions varied widely from one installation to another; the designer needed some means of calculating how big the restrictor had to be and for what length of time it had to operate.

Communications

Professor Herbert Addison, O.B.E., M.Sc., M.I.Mech.E., wrote in connexion with Dr. Krol's paper that the author had stated that the two variables were the length of the impulse valve stroke and the weight of the impulse valve, but it was difficult to find from the paper what were the actual values selected for each of the series of experiments. He asked whether the missing data could be supplied.

The author's conclusion (3) would be still more acceptable if it could be supported by typical figures. He wondered whether experimental results could be selected relating to the conditions of constant supply head, constant delivery head, constant impulse valve weight, and varying impulse valve stroke. If efficiency and rate of discharge were then plotted against impulse valve stroke, it could be seen which setting gave the highest efficiency, and which gave the highest rate of pumping.

In regard to efficiency, the maximum values recorded in the paper hardly seemed to do justice to the merits of the hydraulic ram or to the care which had evidently been devoted to the experimental observations. Whereas the graphs showed no efficiency higher than about 53 per cent, other installations on the same general scale as the author's had yielded values of nearly 70 per cent efficiency. He wondered whether it would be possible to modify the experimental equipment with the aim of improving its performance. The author's conclusion (8) itself offered encouragement. If doubling the size of the air chamber raised the efficiency by 10 per cent, a still further increase in capacity might be profitable. It was difficult to understand why that recorded gain was so consistent. If the chamber had always been kept fully charged with air, its effect might have been expected to vary with the mean pressure within it and with the rate of discharge of the water flowing through it.

A possibility of improvement that other experimenters (for example, Parker 1913)[*] had studied was to control the movement of the impulse valve by a helical spring instead of by a dead load. The more rapid response of the valve probably obtained by doing so might help to reduce the area Q_3 (Fig. 24) and thus lower the energy loss in the system. Doubtless a partial explanation of the low efficiencies quoted in the paper could be revealed if an energy balance were drawn up, showing what percentage of the energy input to the ram was dissipated in the drive pipe, how much was carried away by the water escaping past the impulse valve, and so on.

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[*] Parker, P. A. M. 1913 "The Control of Water as Applied to Irrigation, Power, and Town Water Supply Purposes" (George Routledge and Sons, Ltd., London).

A most valuable part of the paper was its preamble, which traced the historical development of the hydraulic ram. He wondered, however, whether the unavoidable noisiness of the device had not been treated rather too lightly, and whether the author had received complaints about noise from other occupants of the building in which he had made his experiments.

Professor Robert W. Angus, B.A.Sc., M.E., M.I.Mech.E., wrote that the care shown in the experimental work described in the paper by Mr. Binnie and Dr. Thackrah indicated that the results were accurate and might be accepted with confidence. The authors were to be commended for the methods used in obtaining accurate pressure records by the electrical method. He asked if any history of the movement of the valve J was recorded, for, in the short pipe used, the return wave came back to the valve in a small fraction of a second, which was perhaps no longer than the time taken for the valve to close.

The instance where the water column in a pump-discharge line parted, owing to low pressure at some point in it when the pump was cut out, was not uncommon in practice, and might occur in pipes where the pump pressure exceeded about 40 feet of water, and where the axis of the pipe rose quickly at the start and then remained nearly horizontal until the water was discharged into an open reservoir. The problem had been discussed by various writers and he had treated it in several papers*. He had met several instances of the water column parting. The point of rupture of the column was readily located theoretically, and experimental observation confirmed the accuracy of the location; he had studied that problem by the use of the graphical method explained in his papers and had shown how to calculate the maximum distance between the columns, and also the volume of air that would enter through an air valve at the point of separation. The authors' mathematical study of the same problem had added materially to the knowledge of the subject.

The problem of the parted water column must receive careful attention or serious damage might be done when the columns reunited, unless some protection were offered. The idea given in the paper, which dealt in particular with the effect of confining air within the space between the columns and using it as a cushion between the approaching parts of the column, was not new and had often been suggested to him. He had never adopted it, however, because he felt that it was too unreliable. It was an inexpensive method and he hoped that the authors could give examples of where it had been tried. Leakage of the air would result in failure. He knew of an instance where the air inlet on a large main failed to open and damage would have resulted but for other protection. Naturally the paper dealt with a more nearly ideal arrangement than was met in practice.

The authors had named several proposed plans for dealing with the parted column, and had inferred that two of them were ineffective in reducing the pressure due to sudden starting up. They were mistaken in making this criticism of his method, because provision was made for filling the vacant space with water before pumping was restarted. Apparently, however, the authors were referring to general protection against water hammer; if so, he had never, in a long experience with pumps and pumping lines supplied by centrifugal pumps, encountered an instance of dangerous pressure rise when starting up, except where the circuit breaker opened before the pump was up to maximum speed—when the normal water hammer problem arose. Undoubtedly air chambers were a great help, but they appeared not to be the best solution to the problem, and the defects of the spring-loaded relief valve were well known; the authors had rendered a service, however, in obtaining a record of what happened in it.

The method of reducing water hammer by means of a short pipe used as a by-pass around the pump should be further explained, as it did not seem practicable unless the by-pass contained some sort of specially designed valve, which was normally closed but opened for the return flow of water. Flywheels were often surprisingly effective, even in relatively small sizes, and he had not noticed trouble with them in starting up, even with "across the line" motors. In a recent instance a low-head propeller pump had been badly installed in a short line with only a foot valve to check the backward flow of the water. The first time the pump had been stopped the casing had been cracked by water hammer, but the trouble had been entirely cured by attaching a relatively small flywheel to the pump shaft. The necessary dimensions of the flywheel were not difficult to determine.

Mr. R. C. Green (Winchester) wrote that he had been concerned with many hydraulic-ram installations having almost all possible working conditions, and, although the hydraulic ram appeared to be an exceedingly simple machine, his firm did not claim to have mastered all its capabilities.

His firm was careful as to the arrangement of the supply "catchpit" or tank to ensure that the water passing into the drive pipe was absolutely free from air, and, in his opinion, with the arrangement shown in Dr. Krol's paper, owing to the use of a centrifugal pump and the manner in which it was connected to the supply tank, the water would be aerated. He would expect the air vessel to be frequently over-filled with air and the automatic working to be affected.

In his experience the weight of a metal discharge valve had no effect whatsoever on the efficiency of the ram provided it was of sufficient capacity to pass the required quantity of power water. A metal discharge valve would not generally work automatically where the ratio of fall to lift was less than 1/6 unless it were assisted from its seating by either a light spring or rubber buffer.

It was not advisable to use simple types of ram with lifts exceeding 500 feet.

He had long appreciated the superiority of the rubber delivery valve made with large area and lowest possible lift or opening, and, in consequence, he had designed a rubber outlet valve. Such valve was fitted with a single, simple, central adjusting spindle capable of adjusting the varying quantities of power water flowing through the pulse valve. For instance, a 2-inch drive-pipe passing from 10 gal. per min. down to 3 gal. per min., gave uniform efficiency over that range, and for a 3-inch drive-pipe the range was 30 gal. per min. down to about 10 gal. per min.

The rubber outlet valves were found to give 5 per cent better deliveries generally than metal valves. There was no necessity ever to send the valve bodies to the works for reseating, as a new rubber valve could be fitted by any unskilled person of average intelligence.

Rubber valves lasted from three to twenty years according to the working conditions and quality of the water.

Mr. P. Linton, B.Sc., A.M.I.Mech.E., also wrote that he thought the claims of Mr. Binnie and Dr. Thackrah concerning the natural frequency of the pressure gauge used for the tests should not go unchallenged. While they had correctly quoted Wolfe's recommendation for a ratio of 0·3 between the highest frequency of the pressure to be recorded and the natural frequency, the natural frequency referred to the pressure pick-up as a whole and not to the diaphragm itself. A calculation showed that the passages connecting the cavity below the diaphragm to the pipe had a natural frequency of 5,500 cycles per sec. when filled with air, while the cavity itself, considered as a Helmholtz resonater, oscillated at 1,440 cycles per sec.

It had been found in practice† that resonance actually occurred at approximately 4,000 cycles per sec. when the cavity was full of combustion gases. The authors had mentioned that the passages were filled with vaseline, when the natural frequency was likely to be about 14,000 cycles per sec. If, however, some air had been trapped underneath the membrane, resonance would occur at a frequency nearer to that for air. It was probable, therefore, that the maximum frequency component which the gauge could follow accurately, was only about 2,500 cycles per sec.

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* Angus, R. W. 1937 Proc. I.Mech.E., vol. 136, p. 245, "Water Hammer in Pipes, Including those Supplied by Centrifugal Pumps: Graphical Treatment".

1939 Trans. American Soc. Civil Eng., vol. 104, paper No. 2024, "Water Hammer Pressures in Compound and Branched Pipes".

† Private communication from Mr. Wolfe.

Mr. D. G. YOUNG, B.E., and Mr. R. D. WATKINS, B.E. (Brisbane, Australia), wrote in connexion with the paper by Mr. Binnie and Dr. Thackrah that they had recently been engaged on experiments for a thesis on the Electronic Measurement of Water Hammer.

Pressure surges had been measured adjacent to a gate valve at the end of a simple pipe-line connected to an open tank.

Water-hammer Measurement. In the experiments, from tank to valve the line consisted of twenty lengths, each of 21 feet, connected by screwed couplings. The line was straight and free from branches, and was of uniform section except for discontinuities at couplings.

Behind the previously calibrated gate valve was placed a special adapter to carry the pressure head of the measuring instrument. The valve spindle had been modified to permit lever operation for rapid closures.

The pressure head was of capacity-diaphragm type and the pressure diaphragm was arranged as one pole of a parallel-plate condenser. Any variations in pressure caused deflexions of the diaphragm, and hence changes in the capacity of the condenser, causing, in turn, an alteration of the potential across the poles of the condenser. By connexion to a capacity micrometer those changes could be shown on a cathode-ray oscilloscope.

The velocity of flow in the pipe had been estimated from the time taken to drain a portion of it, and, in testing, the valve had been closed, the pipe and tank filled, and the pressure head connected to the micrometer detector and the oscilloscope. The machines had been switched on for about 20 minutes to warm up. The micrometer detector had been set to the required sensitivity, the valve opened, the detector voltmeter set to zero, and the valve closed. The spots on the oscilloscope screen had been adjusted for brightness and focus, and set to the centre of the screen.

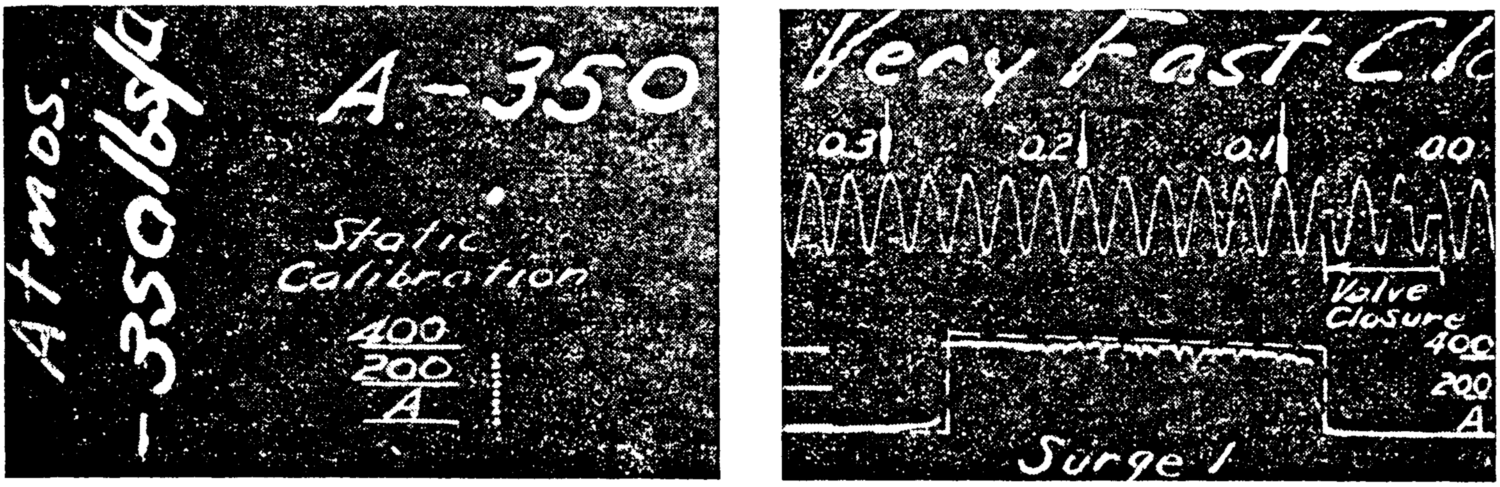
The valve having been opened and the flow having become constant, the valve was closed, as rapidly as desired, and the pressure changes were recorded by a camera.

Comparison of Calculated and Experimental Data. The photographic copies of sections of the recorded trace for the time periods covering the initial pressure surge due to valve closure and the subsequent surges following separation showed reasonable agreement with a theory that makes allowance for losses (Fig. 33). However, owing to the large energy losses assumed to occur at separation, the calculated values might not be regarded as a reliable check on recorded values.

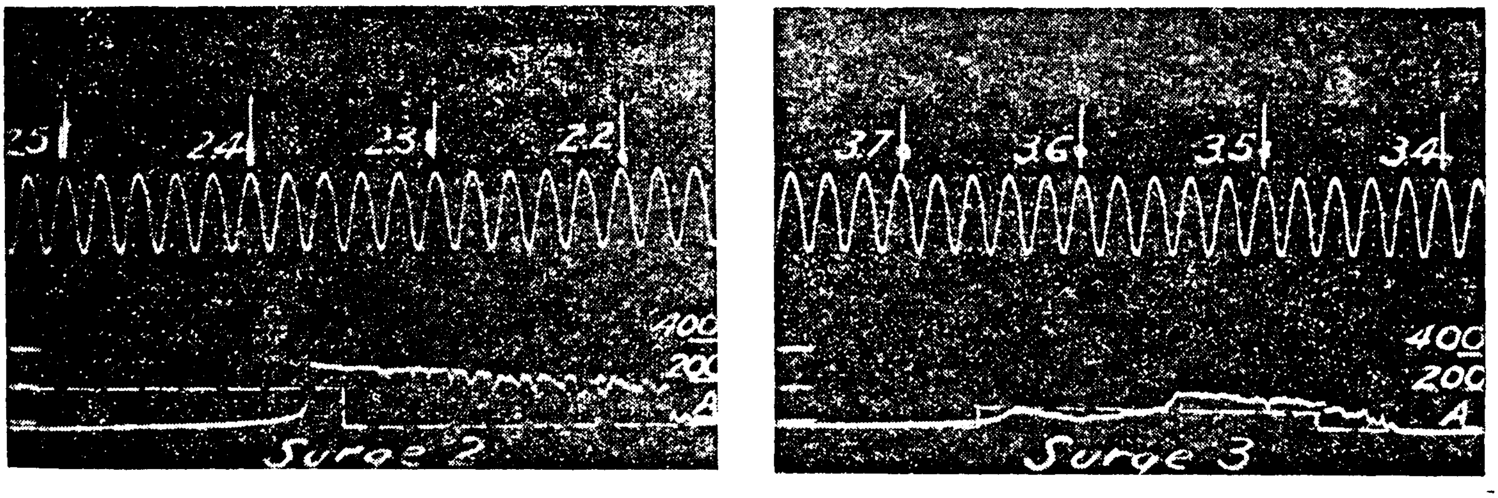
Actual surge records showed two particular characteristics:—

(1) Owing, presumably, to the breakdown of the surface of separation after the initial surges, there was a degeneration of the surges after surges 1 and 2 (Fig. 33*b* and *c*), from a step-type function to one with more gradual rise and fall.

(2) Pressure surges, particularly surges 1 and 2, showed transient variations occurring more or less at uniform intervals of time. It was proposed to investigate those further, with respect to the hypothesis that those effects were due to the discontinuities of joints.



a Static calibration. *b* Pressure surge 1.



c Pressure surge 2. *d* Pressure surge 3.

Fig. 33. Comparison of Calculated and Recorded Surges

Mr. A. M. Binnie and Dr. D. G. Thackrah wrote in reply to the discussion and communications that they regretted that they could not add scales to Mr. Wauchope's Fig. 30 because they did not know the relation between pressure and velocity at the pump outlet while the pump was coming to rest. However, with the aid of the formulae given in the paper they had worked out the maximum rebound pressures in a protected and an unprotected pipe-line for various ratios of the length L to the static lift H. Even that problem was too intricate to admit of a general solution, and it was found necessary to adopt numerical values of the various quantities involved.

The pipe-line was taken to be 12 inches in diameter with a hydraulic gradient of 1 in 50 at an initial velocity $v_0 = 8$ ft. per sec. The air valve was supposed to be unloaded so that $P = p_0 = 34$ feet of water. With a constant static lift $H = 50$ feet the calculations were performed for five values of L: 10, 10^2, 10^3, 10^4, and 10^5 feet, the results being shown in Fig. 34. The assumptions made being different from those of Mr. Wauchope, the absence of a minimum was to be expected. The effect of variations in L is seen to be small when L/H is small because inertia forces predominate over friction forces. At the other extreme where L/H is very great indeed, friction is all important and an air valve is unnecessary. Mr. Wauchope had asked whether the pressure surge would be raised because the pumps did not stop instantaneously. In view of Fig. 5 the authors did not think that a definite and general answer could be given to that question, again because the pressure and velocity changes during the slowing-up period depended upon the details of the plant.

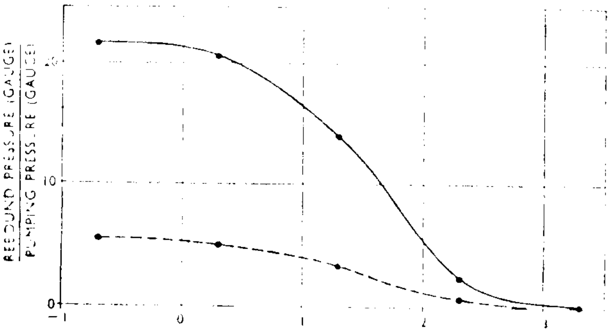


Fig. 34. Effect of Changes in Pipe-line Length upon Maximum Pressures

——— Unprotected. — — — — With air valve.

In regard to the communication by Professor Angus, recently they had seen advertisements of large mechanically operated by-pass valves which were available in the United States. In their experiments they had not tried to measure the valve movements.

Mr. Linton had worked out the natural frequencies of the space below the diaphragm and of the connecting passages when full of air and when full of vaseline. They thought it unlikely that the whole space had been completely filled with vaseline. As Mr. Wolfe had suggested, the vaseline would exert a very marked damping effect (not readily amenable to calculation) on local oscillations in the passages. They had been agreeably surprised to find that the gauge followed the violent initial and final pressure changes without overshooting, therefore they had thought it unnecessary to attempt the difficult task of measuring the natural frequency of the gauge.

The photographic records sent in by Mr. Young and Mr. Watkins were extremely interesting. For the theory of surges in a viscous liquid flowing in a pipe they should consult a recent paper*. There the solution of the differential equation governing the motion was obtained and applied to tests carried out on very long pipe-lines by the Joint Surge Conference†.

Since the paper was read, further experiments with the same apparatus had been carried out by Mr. P. H. D. Hancock. First, an attempt was made to verify the conclusion that the pressure fluctuations in Fig. 14 were due to reflections from bends, sockets, etc. With that end in view, the straight length between fittings f and g, listed on p. 51, was removed together with the bend g. They were replaced by the arrangement shown in plan in Fig. 35, below which the distances measured from the disk valve are given. The new values for h and the total length L were 168·8 and 181·7 feet, and the friction coefficient k was increased to 0·414. Observations of the pressure rise were made as before at $v_0 = 9$ ft. per sec.

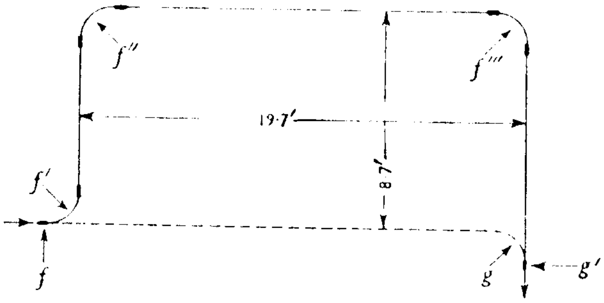


Fig. 35. Alterations to Pipe-line

f' At 111·4 feet, a 90 deg. bend.
f'' At 119·9 feet, a 90 deg. bend.
f''' At 139·4 feet, a 90 deg. bend.
g' At 148·8 feet, a socket.

The resulting diagram, which was closely repeatable, is shown in Fig. 36, which should be compared with Fig. 14. As far as f the trace was nearly, although not quite, unaltered, but the remainder differed widely. Exact analysis was impracticable because a complex system of secondary reflections was set up in the pipe-line, but there could now be little doubt that the fluctuations were, in fact, due to reflected waves.

In the course of the work it was found on one occasion that the expected water hammer did not occur. Its disappearance was traced to the accidental entry of air, due to the failure of a gland packing on the centrifugal pump that supplied the pipe-line. The defect was immediately rectified, but it had suggested that another method of eliminating violent shocks was to admit a small and continuous stream of air, thus providing a cushion, not concentrated near the valve but distributed uniformly along the pipe-line. Accordingly, an air admission valve was substituted for the suction gauge on the pump, and the air that passed through the pipe-line was trapped by a bell in the reservoir and measured by a gas meter. It was important to ensure that no air lodged permanently in the pipe-line, therefore all the experiments were carried out at $v_0 = 9$ ft. per sec., which was the velocity normally used to clear the pipe-line of air.

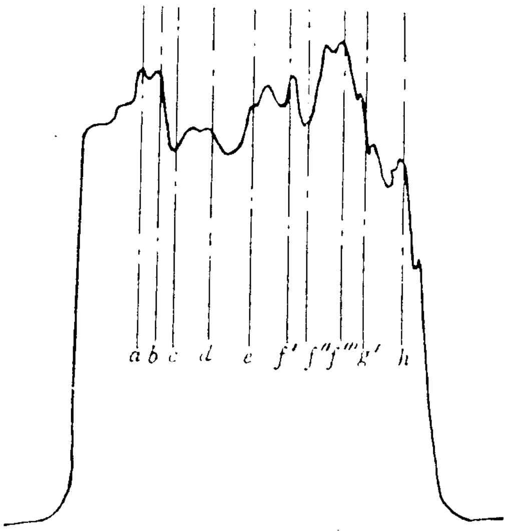


Fig. 36. Tracing of First Pressure Rise in Modified Pipe-line $v_0 = 9$ ft. per sec.

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* Binnie, A. M. 1951 *Quarterly Journal of Mechanics and Applied Mathematics*, vol. 4, p. 330.

† Final Report (1949) of the Joint Surge Conference: Middle East Pipelines, Ltd., 293 Regent Street, London, W.1.

As a further check, the following tests were made. When the air valve had been open for a considerable time, it was suddenly shut; and after the lapse of 33 seconds the air flow at the reservoir end was seen to cease. When the valve was opened, the first bubbles appeared in the reservoir after 33 seconds, the full air flow being restored about 10 seconds later. In a repetition of the latter test, strong potassium permanganate solution was drawn in instead of air, the corresponding time being 31 seconds. Those observations were deemed satisfactory. The admission of even a minute stream of air completely suppressed audible water hammer, therefore the electronic gauge was replaced by the engine indicator previously used. The resulting diagrams showed a succession of surges of rapidly diminishing magnitude. The maximum pressures that were recorded with various air admissions are shown in Fig. 37, which demonstrate how effective that simple method is; a comparison with Fig. 15*a* is reasonable because the small increase in the length of the pipe-line could have had little influence upon the maximum pressure developed. The reduction in the efficiency of the pump due to the air was not known, but the method is clearly worth consideration in circumstances where no objection could be raised to the presence of a slight amount of air in the pipe-line.

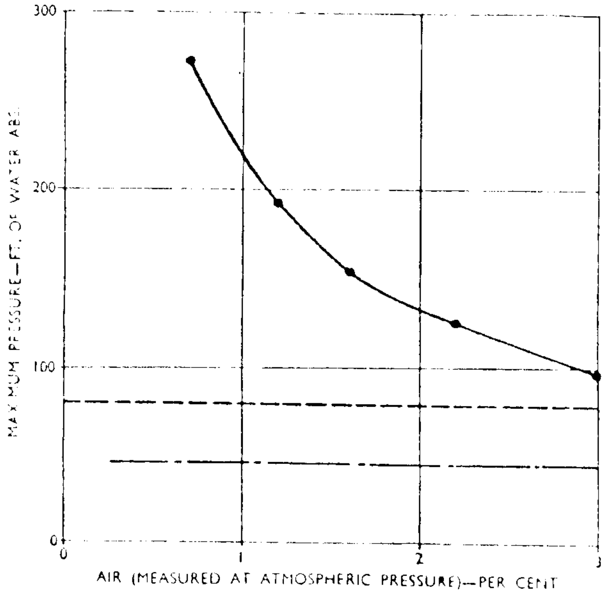

Fig. 37. Maximum Pressures with Continuous Air Admission

— — — — Normal pumping head. ——— - ——— Normal static head.

Dr. J. KROL wrote in reply that he appreciated Mr. Wauchope's remark that the paper might lead to other developments on the subject, as that was exactly what he had in mind when he undertook the research. Although the automatic hydraulic ram appeared at first to be an exceedingly simple machine, theoretically it involved a number of complex phenomena such as unsteady flow of water, water hammer, and the automatic motion of the impulse valve. The principal objects of the research were (1) to separate the important factors affecting the performance of the machine, and (2) to find mathematical relationships between them with a view to facilitating an intelligent design. The need of such work was substantiated by Mr. R. C. Green's remarks.

The paper was limited to consideration of a simple automatic hydraulic ram with a rigid impulse valve. Future research workers might find an interesting field of investigation in the study of a similar ram, with particular reference to the effects of various shapes of (1) rigid impulse valves, expressed as functions $\Phi(s)$ and $R(s)$, (2) spring-loaded impulse valves, (3) resilient impulse valves, and (4) turbine-like impulse valves. After those fundamental studies were completed, further investigations could be made on the development of the numerous variations of the hydraulic ram mentioned in the paper; in particular, a hydraulic ram installation designed for large quantities of water and involving the application of a concrete drive pipe with a turbine-like impulse valve.

Mr. H. Green's remark that the weight of the impulse valve did not affect the efficiency appeared to be at complete variance with his own experience, based upon careful analysis of the behaviour of impulse valves weighing 0·605, 1·485, 2·360, 3·245, 4·125, and 5·005 lb. respectively. Possibly the type of impulse valve used by Mr. Green had such characteristics that no marked effects of the load variations were noticeable; but it was a specific and not a general case. The effect of the weight of the impulse valve was studied in connexion with the determination of the coefficient of drag, expressed by formula (38). At reduced lengths of the impulse valve stroke, higher loads could be used. Thus, for example, up to $\frac{1}{8}$ inch length of impulse valve stroke, the ram would operate with the heaviest weight of 5·005 lb. When the length of the impulse valve stroke was increased up to $\frac{3}{16}$, $\frac{1}{4}$, $\frac{5}{16}$, and $\frac{3}{8}$ inch, the maximum impulse valve weights were 4·125, 3·245, 2·365, and 1·485 lb. respectively. Those experimental results were further confirmed by analytical studies leading to formulae (50) and (51).

In regard to the questions raised by Professor Addison, in the experimental verification of the theory developed, the characteristics of the machine were varied by means of twenty-six combinations of the length of the impulse valve stroke *s* and its weight *W*. Those were listed in Table 1.

TABLE 1

Series No.	*s*, inch	*W*, lb.	Series No.	*s*, inch	*W*, lb.
1	$\frac{1}{16}$	0·605	14	$\frac{3}{16}$	1·485
2	$\frac{1}{16}$	1·485	15	$\frac{3}{16}$	2·365
3	$\frac{1}{16}$	2·365	16	$\frac{3}{16}$	3·245
4	$\frac{1}{16}$	3·245	17	$\frac{3}{16}$	4·125
5	$\frac{1}{16}$	4·125	18	$\frac{1}{4}$	0·605
6	$\frac{1}{16}$	5·005	19	$\frac{1}{4}$	1·485
7	$\frac{1}{8}$	0·605	20	$\frac{1}{4}$	2·365
8	$\frac{1}{8}$	1·485	21	$\frac{1}{4}$	3·245
9	$\frac{1}{8}$	2·365	22	$\frac{5}{16}$	0·605
10	$\frac{1}{8}$	3·245	23	$\frac{5}{16}$	1·485
11	$\frac{1}{8}$	4·125	24	$\frac{5}{16}$	2·365
12	$\frac{1}{8}$	5·005	25	$\frac{3}{8}$	0·605
13	$\frac{3}{16}$	0·605	26	$\frac{3}{8}$	1·485

In regard to conclusion (3), from Fig. 28*a* for the delivery head of approximately 82 feet, the highest efficiency of about 0·53 could be obtained by the use of either impulse valve adjustment denoted as series 4 or series 5. Series 4 gave the power of about 2·75 ft.-lb. per sec., whilst the corresponding result for series 5 was about 3·40 ft.-lb. per sec. In those circumstances, series 5 was surely more advantageous. Similarly, from the results given by series 13 and 16, shown in Fig. 28*b*, for a delivery head of approximately 87 feet, both series gave the same efficiency of about 0·46, whilst the power produced by the series 13 was about 3·80 ft.-lb. per sec. as compared with about 11·8 ft.-lb. per sec. given by series 16.

He agreed with Professor Addison that maximum values for efficiency recorded in his paper were low. The reasons for those low efficiencies were (1) the use of an old drive pipe with the factor $4f$ estimated to be in the region of 0·0675, and (2) a comparatively high supply head, $H = 13{\cdot}0$ feet. The main object of the research presented was to find a satisfactory correlation between the theory and experiment to facilitate the solution of design problems and further development studies.

The theoretical analysis of the effect of the volume of the air chamber was very complex. The information available seemed to indicate that the air chamber should be designed to contain a volume of compressed air equal to approximately 100 times the volume of water delivered per cycle. Also, the level of water contained in the air chamber should be as low as possible in relation to the supply head H. Those requirements called for a short air chamber of large diameter.

The noise caused by the operation of the hydraulic ram was closely connected with the application of the rigid impulse valve. Further research relating to the application of non-rigid impulse valves was likely to eliminate that nuisance. In particular, three types of non-rigid impulse valves should be considered (1) spring-loaded, (2) resilient, and (3) turbine-like.

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AN ANALYTICAL AND EXPERIMENTAL STUDY OF THE HYDRAULIC RAM

BY

WALLACE M. LANSFORD

AND

WARREN G. DUGAN

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AN ANALYTICAL AND EXPERIMENTAL STUDY OF THE HYDRAULIC RAM

I. INTRODUCTION

1. *Preliminary Statement.*—The hydraulic ram is a machine which uses the kinetic energy in a moving column of water to lift part of that water to a height greater than that of the source of supply. With no external addition of energy, a well-made hydraulic ram will pump a part of the water with which it is supplied to any height from about twice the supply head to many times the supply head, with an efficiency which exceeds 65 per cent or even 70 per cent in some ranges of capacity and pressure.

A patent on the hydraulic ram was issued in 1797, to Montgolfier[1]* in France. Since that time, its dependability, simplicity, and economy have led to its frequent use, particularly in isolated districts where no water mains are available. Its application varies from individual installations supplying families or livestock, to larger systems supplying water to small cities.

Despite the widespread use of the ram, the analysis of its actual operating characteristics is not generally known, though empirical performance guarantees are of course made by manufacturers. Although several investigators have studied the problem of a rational analysis, only O'Brien and Gosline[2] may be said to have arrived at a solution. Their solution applies only to rams having a very rigid or stiff waste valve.

2. *Purpose of Investigation.*—It was the purpose of the present investigation to make a rational mathematical analysis of the operation of single-acting automatic hydraulic rams, and to compare the results of such an analysis with those found from an experimental investigation of the rams.

The rams used in this investigation differed considerably in construction[3] and performance from the one used by O'Brien and Gosline. The analysis developed by O'Brien and Gosline neglected the effects of elasticity of the drive pipe and waste valve parts, while in the rams used in this investigation, the effects of elasticity of the resilient waste valve disc were often more marked than the effects of the elasticity of the water. These differences in construction and performance seemed to justify further investigation.

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*Numerical indices refer to correspondingly numbered references in the bibliography at the end of the bulletin.

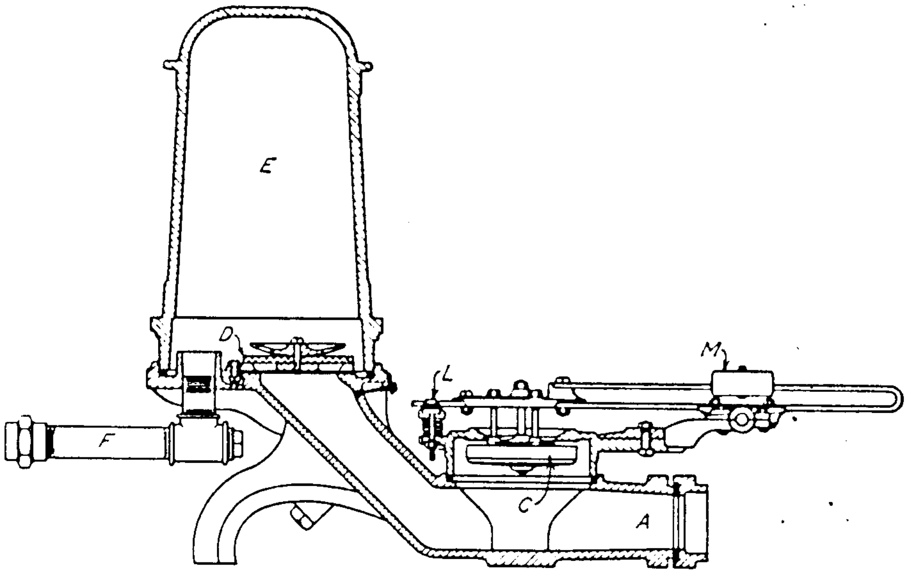


FIG. 1. CROSS-SECTIONAL DIAGRAM OF A HYDRAULIC RAM

3. *Acknowledgment.*—The investigation herein presented was started as a thesis by the second-named author under the supervision of the first-named author. The investigation was continued as a part of the work of the Engineering Experiment Station, of which DEAN M. L. ENGER is the director, and of the Department of Theoretical and Applied Mechanics, of which PROFESSOR F. B. SEELY is the head.

During the investigation pressure-time diagrams for the ram were obtained by use of an electrical pressure gage and an oscillograph which was adapted for use with the ram and operated by the staff of the "Stresses in Railroad Track Investigation." The authors wish to express their appreciation especially to Mr. R. Ferguson, Mr. H. C. Roberts, and Mr. T. P. DeWan of this investigation for the interest shown and help given in operating the apparatus as applied to the ram.

4. *Explanation of Operation of Ram.*—In Fig. 1 is shown a diagram of the valve box of a hydraulic ram manufactured by the Rife Hydraulic Manufacturing Company which will help make clear the operation of the ram. [At *A* is attached the drive pipe, the other end of which is connected to the water supply. If the waste valve *C* is open, as shown in the diagram, water flows into the valve box, then out around the waste valve disc *C*, and is wasted.

As the flow increases, a frictional resistance is developed in the waste valve passage which increases the pressure under the valve disc *C*. When the pressure in the valve box is large enough, the valve and rocker-arm on which it is mounted rise, thereby closing the valve quickly. Thus the flow through the waste valve is stopped, but since the column of water in the drive pipe has a considerable velocity, a very high pressure would be created in the valve box if relief were not provided ② The relief is found in the opening of the check valve *D*, which permits the flow of the water to continue by passing into the surge tank *E*; this tank is filled partly with water and partly with air. The pipe at *F* is connected to the delivery tank, so that the pressure in the surge tank *E* is the delivery pressure. Water continues to flow into the surge tank until the higher pressure which exists there reduces the velocity to zero ③ The check valve then closes and the waste valve opens, completing the cycle. The duration of the cycle is usually from one-half to one and one-half seconds. The surge tank is large enough to cause the flow from it to be sensibly steady, though the pumping into it is intermittent.

5. *Objectives of Analysis.*—Many variables are involved in the operation just described, but the ones of primary interest to the designing or operating engineer are the rate of pumping and the rate of wasting (weight of water pumped or wasted per unit of time). Therefore the objective of the analysis herein presented was to determine the rate of pumping and of wasting for any condition of operation, having previously obtained the physical dimensions and the experimental constants of the apparatus for that condition of operation. A list of constants which are necessary for the analysis is given in the last column in Table 1 (page 36). The definitions of the symbols for the constants are given on page 12.

II. ANALYSIS OF ACTION OF RAM

6. *Method of Approach.*—The problem is attacked by attempting to obtain the relation between velocity and time for the water column in the drive pipe during each part of the cycle. From this relationship the quantity of water wasted and the quantity pumped per cycle may be found; the time elapsed during a cycle may also be determined. From these values the time rate of pumping and of wasting may be easily determined.

7. *Division of Cycle into Periods.*—It was necessary to divide the cycle into its separate parts rather precisely, and to analyze each

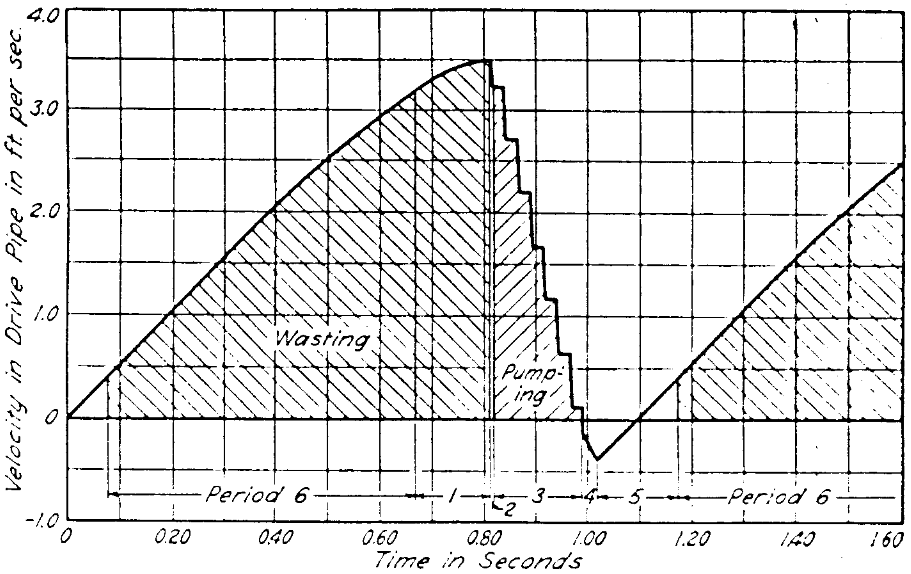


FIG. 2. RELATION OF VELOCITY IN DRIVE PIPE AT RAM TO TIME

part individually. It seemed logical to divide the cycle into six separate periods, during each of which the velocity is affected by different factors and therefore varies according to a different law. Reference to Fig. 2, which is a diagram of the assumed relation of velocity in the drive pipe to time, will clarify the six periods. In the analysis of the ram the drive pipe was assumed to be horizontal.

The cycle was considered to start at the instant of the beginning of waste valve closure. This was a logical point in the cycle at which to begin the analysis, since, from a consideration of the factors controling the operation of the waste valve, it seemed reasonable to assume that the velocity of the water in the drive pipe at this instant is independent of supply pressure and delivery pressure, and depends only on the setting of the waste valve. At this time water is being wasted through the open waste valve, and the wasting continues until the valve is completely closed. The first period includes the time from the instant the waste valve begins its motion to the instant at which it is completely closed. That is, Period 1 includes the time during which the waste valve closes.

Immediately after the waste valve closes, the pressure in the valve box is only slightly above atmospheric, but the water, traveling at a considerable velocity, compresses itself, expands the pipe, and compresses the rubber waste valve disc, as the pressure rises. When

the pressure in the valve box becomes equal to the delivery pressure or slightly greater, the check valve opens. The distance through which the check valve moves is comparatively small and for convenience the motion of the check valve is assumed to be instantaneous. The second period includes the time between the instant of complete closure of the waste valve and the instant of opening of the check valve.

Flow into the surge tank now occurs, and continues until the unbalanced force caused by the difference between delivery and supply pressures reduces the velocity through the check valve to zero. The third period includes the time during which the check valve is open.

The check valve now closes, and the water, under delivery pressure at the valve box, but under supply pressure at the other end of the drive pipe, is accelerated away from the ram, toward the supply, acquiring a "negative" velocity. The displacement due to this negative velocity lowers the pressure in the valve box until the waste valve will no longer remain closed. The fourth period includes the time between the closing of the check valve and the beginning of opening of the waste valve.

The waste valve now being open, the pressure in the valve box is atmospheric, while at the other end of the drive pipe it is still the supply pressure, assuming a horizontal drive pipe. However, the negative velocity, toward the supply, still persists, partially emptying the valve box, until the force caused by the supply pressure retards the velocity of the water column, stops the motion, and finally gives the water column a velocity directed back toward the ram, refilling the valve box. The fifth period is chosen to include the time between the beginning of opening of the waste valve, and the instant the valve box is refilled and wasting begins.

Water continues to waste through the open waste valve until the difference in pressure on the two sides of the valve is enough to overcome its weight, when the waste valve begins to close. The sixth period includes the time from the beginning of wasting to the beginning of waste valve closure. This completes the cycle.

The foregoing division of the cycle into periods may be briefly summarized as follows:

Period 1 includes the time from the instant the waste valve begins to close until the instant at which it is completely closed.

Period 2 includes the time between the instant of complete closure of the waste valve and the instant of opening of the check valve.

Period 3 includes the time during which the check valve is open.

Period 4 includes the time between the closing of the check valve and the beginning of opening of the waste valve.

Period 5 includes the time between the beginning of opening of the waste valve and the instant when wasting begins.

Period 6 includes the time from the beginning of wasting to the beginning of waste valve closure.

This separation into periods is the basis for the analysis of the operation of the ram.

8. *Definition of Notation.*—

Symbol	Definition	Dimensions
A =	Cross-sectional area of drive pipe	ft.^{2}
A_v =	Area of waste valve disc	ft.^{2}
a =	Velocity of pressure-wave transmission in drive pipe	ft./sec.
B =	Area on velocity-time diagram of Fig. 4	ft.
B_r =	Area on velocity-time diagram of Fig. 6	ft.
b =	Friction coefficient of drive pipe	dimensionless
c =	Friction coefficient of waste valve	dimensionless
c_1 =	Constant of integration	
c_2 =	Constant of integration	
E =	Modulus of elasticity of drive pipe material	lb./ft.^{2}
E_v =	Average stiffness modulus of waste valve disc	lb./ft.
e =	Base of natural logarithms, = 2.718	dimensionless
g =	Acceleration of gravity	ft./sec.^{2}
H =	Static supply head	ft.
h =	Static delivery head	ft.
h' =	Instantaneous head in valve box	ft.
h_0' =	Magnitude of pressure wave developed during Period 2	ft.
h_0'' =	Magnitude of pressure wave developed during second surge of Period 3	ft.
h_0 =	Average magnitude of pressure waves developed during Periods 2 and 3	ft.
J =	Acceleration of waste valve during Period 1, assumed constant	ft./sec.^{2}
j =	Combined friction constant, $= b + c + 1$	dimensionless
K =	Volume modulus of elasticity of water	lb./ft.^{2}
k =	Time rate of decrease of acceleration of water column in drive pipe during Period 1, assumed constant	ft./sec.^{3}
L =	Length of drive pipe, supply tank to center of waste valve	ft.
L_1 =	Length of drive pipe, supply tank to center of check valve	ft.
l =	Length of water column in drive pipe	ft.

Symbol	Definition	Dimensions
m =	Frictional constant of check valve, so that head loss $= \frac{mv}{2g}$	ft./sec.
N =	Number of surges in Periods 2 and 3	dimensionless
p =	Functional symbol indicating presence of physical constants in a function	
Q =	Rate at which water is wasted	lb./min.
Q_1 =	Water wasted during Period 1	lb./cycle
Q_6 =	Water wasted during Period 6	lb./cycle
Q_w =	Total water wasted	lb./cycle
q =	Rate at which water is pumped	lb./min.
q_p =	Water pumped	lb./cycle
r =	Radius of drive pipe	ft.
S_v =	Length of stroke of waste valve	ft.
s =	Value of the displacement of the waste valve from its closed position	ft.
t =	Time	sec.
t_1 =	Duration of Period 1	sec.
t_2 =	Duration of Period 2	sec.
t_3 =	Duration of Period 3	sec.
t_4 =	Duration of Period 4	sec.
t_5 =	Duration of Period 5	sec.
t_6 =	Duration of Period 6	sec.
t_c =	Duration of one cycle	sec.
t' =	$\frac{2L_1}{a} - t_2$	sec.
t_r =	Duration of last portion of Period 3, see Fig. 3	sec.
u =	Thickness of wall of drive pipe	ft.
v =	Velocity of water in drive pipe	ft./sec.
v_0 =	Value of v when waste valve begins to close	ft./sec.
v_1 =	Value of v at end of Period 1	ft./sec.
v_2 =	Value of v at end of Period 2	ft./sec.
v_3 =	Value of v at end of Period 3	ft./sec.
v_4 =	Value of v at end of Period 4	ft./sec.
v_5 =	Value of v at end of Period 5	ft./sec.
v_6 =	Value of v at end of Period 6, $= v_0$, since cycle is repeated	ft./sec.
v_r =	Value of v near end of Period 3, see Fig. 3	ft./sec.
Δv =	Reduction in velocity required to produce pressure head h_0	ft./sec.
$(\Delta v)_1$ =	Reduction in velocity required to produce pressure head h_0'	ft./sec.
$(\Delta v)_2$ =	Reduction in velocity required to produce pressure head h_0''	ft./sec.
w =	Specific weight of water	lb./ft.^{3}

Symbol	Definition	Dimensions
Y	= Ratio of change of pressure head in valve box to volume change of valve box $= \dfrac{E_v}{wA_v^2}$	1/ft.^{2}
Z	$= \dfrac{a}{AgY}$	sec.
α_6	= Acceleration of water column in drive pipe at end of Period 6 or beginning of Period 1	ft./sec.^{2}
σ	= Poisson's ratio for drive pipe material	dimensionless

9. *Analysis of Period 1.*—Attempts at rational analysis of the action during this period were fruitless. Some approximations were therefore made.

Records of the actual motion of the waste valve were made during many of the pumping tests, as described later in Section 17. A study of these records showed that the acceleration of the waste valve may be approximated by an equation of the form

$$\frac{d^2s}{dt^2} = -J \tag{1}$$

where s is the value at any instant of the displacement of the waste valve from its closed position, and J is the acceleration of the waste valve, which is assumed constant. Since s is measured away from the valve seat, the negative sign must be used to obtain a positive value for J. The evaluation of the quantity J is described in Section 17.

Integration of Equation (1) gives

$$\frac{ds}{dt} = -Jt + c_1 \tag{2}$$

in which c_1 is a constant of integration. But since the valve starts from rest, $\dfrac{ds}{dt} = 0$ when $t = 0$; hence $c_1 = 0$.

Integration of Equation (2) gives

$$s = -\frac{Jt^2}{2} + c_2. \tag{3}$$

But $s = S_0$ when $t = 0$; hence $c_2 = S_0$. Also $s = 0$ when $t = t_1$. Therefore

$$t_1 = \sqrt{\frac{2S_0}{J}}. \tag{4}$$

An approximation to the variation of velocity in the drive pipe is also necessary. It appears from the indicated pressure–time diagrams, shown in Fig. 16 and described in detail in Section 18, that the pressure in the valve box rises very little until the waste valve is almost completely closed; therefore the velocity of the water column in the drive pipe must continue to increase to some extent during Period 1. On the other hand, the friction through the waste valve undoubtedly increases as the valve opening becomes smaller, and the acceleration of the water column in the drive pipe must decrease due to the increased friction loss and to the force required to accelerate the waste valve. The assumption is made that the acceleration decreases at a constant rate, k, with respect to time, and becomes zero at the instant of valve closure. That is,

$$\frac{d^2v}{dt^2} = -k. \tag{5}$$

Integration of Equation (5) gives

$$\frac{dv}{dt} = -kt + c_1.$$

But $\dfrac{dv}{dt} = \alpha_6$ when $t = 0$; hence $c_1 = \alpha_6$. Also $\dfrac{dv}{dt} = 0$ when $t = t_1$; hence $k = \dfrac{\alpha_6}{t_1}$. Thus,

$$\frac{dv}{dt} = \alpha_6 - \frac{\alpha_6 t}{t_1}. \tag{6}$$

Integration of Equation (6) gives

$$v = \alpha_6 t - \frac{\alpha_6 t^2}{2t_1} + c_2. \tag{7}$$

But $v = v_0$ when $t = 0$; hence $c_2 = v_0$. Also $v = v_1$ when $t = t_1$.

$$\text{Then } v_1 = v_0 + \frac{\alpha_6 t_1}{2}. \tag{8}$$

The velocity, v_0, the minimum velocity at which the waste valve will begin to close, is a function of the shape of valve and valve box, the length of the stroke, and the weight of the waste valve. These factors are not easily rationalized. Therefore, v_0 has been determined experimentally.

Since the value of v_0 is known for any setting of the waste valve, and α_6 and t_1 may be determined from Equation (57) and Equation (4) respectively, v_1 may be calculated by means of Equation (8).

The water wasted during Period 1 is

$$Q_1 = \int_0^{t_1} wAv\,dt. \tag{9}$$

Substitution of the value of v from Equation (7) gives

$$Q_1 = \int_0^{t_1} Aw\left(v_0 + \alpha_6 t - \frac{\alpha_6 t^2}{2t_1}\right)dt = wA\left(v_0 t_1 + \frac{\alpha_6 t_1^2}{3}\right). \tag{10}$$

From Equation (10) Q_1, the water wasted during Period 1, may be calculated.

At the end of Period 1, then, the water column in the drive pipe is traveling at the known velocity v_1, and has wasted an amount Q_1 in a time t_1. Both waste valve and check valve are closed. Since the acceleration is assumed zero, and pipe friction disregarded, the pressure head is $H - v_1^2/2g$ at all points in the pipe. The velocity head is small compared with the supply head (from 0.4 per cent to 3.7 per cent of the supply head for the tests herein reported), and is even smaller compared with the delivery head. Therefore the velocity head is disregarded in the analysis which follows. That is, the pressure head at all points in the drive pipe is assumed to be H at the end of Period 1.

10. *Analysis of Period 2.*—Continued flow in the drive pipe compresses the waste valve disc, expands the pipe, and compresses the water; these displacements accompany a rise of pressure. According to fundamental water hammer theory, this pressure rise is transmitted back along the drive pipe at the velocity of pressure wave propagation, denoted by a. Further, the velocity of that part of the water column in which the pressure has risen is reduced in direct proportion to the height above H of the pressure head at that instant.[3] That is,

$$\text{reduction in velocity} = \frac{(h' - H)\,g}{a} \tag{11}$$

and if v be now considered as the velocity of the water column at the check valve

$$v = v_1 - \frac{(h' - H)\,g}{a}. \tag{12}$$

The assumption was made that the load-deflection curve for the waste valve disc (see Fig. 7, assumption I) is a straight line, having a slope E_v which is defined as the modulus of the disc. The justification of this assumption is discussed in Section 12. If this assumption is true, a linear relation exists between the change of pressure head in the valve box and the change in volume of the valve box due to the compression of the waste valve disc. That is,

$$\text{change in volume of valve box} = \frac{\Delta h'}{Y}$$

where

$$Y = \frac{\text{Pressure head change in valve box}}{\text{Volume change of valve box}} = \frac{\Delta h'}{\dfrac{A_v w \Delta h'}{E_v} A_v} = \frac{E_v}{wA_v^2}. \tag{13}$$

The change in volume of the valve box must equal the flow into it; that is, for a differential time dt,

$$\frac{dh'}{Y} = Av\,dt. \tag{14}$$

At the end of Period 2, $v = v_2$, and $(h' - H) = h_0'$, where h_0' is defined as the difference between the head required to force water through the check valve at velocity v_2 and the supply head. Substitution of these values in Equation (12) gives

$$v_2 = v_1 - \frac{h_0'g}{a}. \tag{15}$$

From Equation (15) v_2, the velocity at the end of Period 2, may be computed.

Differentiation of Equation (12) gives

$$dh' = -\frac{a}{g}\,dv. \tag{16}$$

Substitution for dh' from Equation (14) gives

$$AvYdt = -\frac{a}{g}\,dv$$

whence

$$dt = -\frac{a}{AgY}\,\frac{dv}{v} = -Z\,\frac{dv}{v} \tag{17}$$

where

$$Z = \frac{a}{AgY} = \frac{awA_v^2}{AgE_v}. \tag{18}$$

Integration of Equation (17) gives

$$t = -Z\log_e v + c_1.$$

But when $t = 0$, $v = v_1$; hence $c_1 = Z\log_e v_1$. Therefore

$$t = Z\log_e\frac{v_1}{v}. \tag{19}$$

This relation of velocity to time continues until the pressure rises enough to open the check valve; at this time the velocity is v_2. Hence the duration of Period 2 is

$$t_2 = Z\log_e\frac{v_1}{v_2}. \tag{20}$$

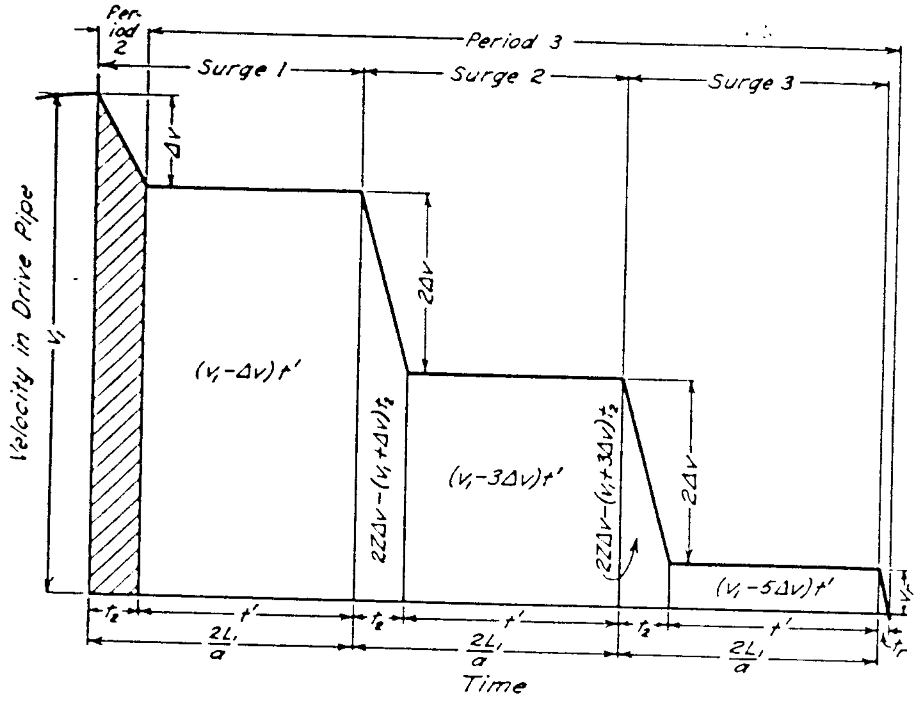


FIG. 3. RELATION OF VELOCITY IN DRIVE PIPE AT RAM TO TIME DURING PERIODS 2 AND 3

It will be noted that the analysis disregards pipe friction during this period. The pressure changes due to pipe friction in a well-installed ram are not important compared to the other pressure variations in this period.

At the end of Period 2, then, the pressure head in the valve box is $h_0' + H$, and the velocity at the check valve is v_2. Beyond a section $a \times t_2$ feet from the check valve measured along the drive pipe toward the supply end, the pressure head H, and velocity v_1, are unchanged from their values at the beginning of Period 2. From the valve box to this section the velocity increases and the rise in pressure over the pressure at the beginning of Period 2 decreases, according to a logarithmic relation.

11. *Analysis of Period 3.*—The analysis of Period 3 extends the work of O'Brien and Gosline to take into account the effects of the elasticity of the waste valve. Reference to Fig. 3, which is a diagram of the assumed relation of velocity at the check valve to time during Periods 2 and 3 will clarify the analysis of these periods. At the beginning of Period 3 the check valve opens, and under the con-

stant differential of pressure head $(h_0' + H) - h$, water is discharged into the surge tank, at a velocity $v_2 = v_1 - (\Delta v)_1$ according to Equation (15) with $(\Delta v)_1$ substituted for $\frac{h_0' g}{a}$. However, in the drive pipe on the supply side of the pressure wave, the velocity is still v_1. As the pressure wave passes any section, the velocity at that section is reduced to $v_1 - (\Delta v)_1$, and the pressure head increased by h_0'. The time required for this pressure rise and velocity decrease to take place at any section is necessarily equal to t_2.

When the first portion of the high-pressure wave reaches the supply end of the drive pipe, the velocity in the drive pipe is $v_1 - (\Delta v)_1$, and the pressure head has been correspondingly increased by an amount h_0', except in that part of the drive pipe at the supply end (of length $a \times t_2$) in which the pressure is still rising. But this higher pressure cannot exist in contact with the constant-pressure supply. Therefore, as the pressure wave reaches the supply end of the pipe, it is reflected back as a "relief" wave, identical in form with the first wave, which reduces the pressure head again by h_0' to its original value H. However, such relief of pressure must reduce the velocity again by $(\Delta v)_1$. Therefore, while the relief wave is traveling back to the ram, the velocity of the water column on the supply side of the wave is $v_1 - 2(\Delta v)_1$ while at the check valve it is still $v_1 - (\Delta v)_1$.

At a time $\frac{2L_1}{a}$ after the beginning of Period 2, the first portion of the relief wave reaches the check valve again, lowering the pressure. The velocity of the water column is still toward the ram, however, and discharge must therefore continue through the check valve. Thus during a length of time t_2, as the first pressure wave is dissipated, another is generated, of magnitude h_0'' which is slightly lower than h_0' because of the reduced friction loss through the check valve at the lower velocity. The generation of the new pressure wave reduces the velocity to $v_1 - 2(\Delta v)_1 - (\Delta v)_2$ in which $(\Delta v)_2$ is less than $(\Delta v)_1$ in the same proportion as h_0'' is less than h_0', as may be seen from Equation (11), by successively substituting $(\Delta v)_1$ and $(\Delta v)_2$ for the left hand member and h_0' and h_0'' for the quantity $(h' - H)$.

The new pressure wave or surge travels back to the supply end of the drive pipe, flow through the check valve meanwhile taking place at the rate of $v_1 - 2(\Delta v)_1 - (\Delta v)_2$. Upon reflection of the second pressure wave as a relief wave, the velocity of the water on the supply side of the wave becomes $v_1 - 2(\Delta v)_1 - 2(\Delta v)_2$, while on the ram side of the wave the velocity is still $v_1 - 2(\Delta v)_1 - (\Delta v)_2$.

This action of one pressure wave succeeding another continues until at last the velocity which remains after dissipation of a pressure wave is insufficient to build up the next pressure wave to a value equal to the delivery pressure. Then the pumping ceases, and Period 3 is completed.

For the sake of convenience in the mathematical treatment of the action in Period 3, the following simplification is made. From Equation (15) and the definition of $(\Delta v)_1$ it is seen that

$$v_1 - v_2 = (\Delta v)_1 = \frac{h_0' g}{a}.$$

The rise in pressure head h_0' necessary to open the check valve is slightly greater than $(h - H)$, due to the friction through the check valve. Therefore h_0' is the sum of the increases in pressure head due to the delivery pressure and the friction loss. That is,

$$h_0' = h - H + \frac{m\,[v_1 - (\Delta v)_1]}{2g} \qquad (21)$$

and

$$(\Delta v)_1 = \frac{g}{a}\left\{(h - H) + \frac{m\,[v_1 - (\Delta v)_1]}{2g}\right\}.$$

Subsequent pressure surges in Period 3 are consecutively less in magnitude, since after each surge the velocity through the check valve is less than it was during the preceding surge. When Period 3 is nearly completed, only a small velocity (see Fig. 3) remains; the friction loss through the check valve is small, and therefore the last pressure surge is not much greater than $h - H$, and the velocity reduction due to this pressure surge is nearly equal to $(h - H)\frac{g}{a}$.

For simplicity, the assumption is made that all pressure surges and velocity reductions from beginning to end of Period 3 are equal in magnitude; this magnitude h_0 is taken as the average between the first surge, h_0', and the approximate value of the last surge, $h - H$. Thus

$$h_0 = (h - H) + \frac{m\,(v_1 - \Delta v)}{4g}.$$

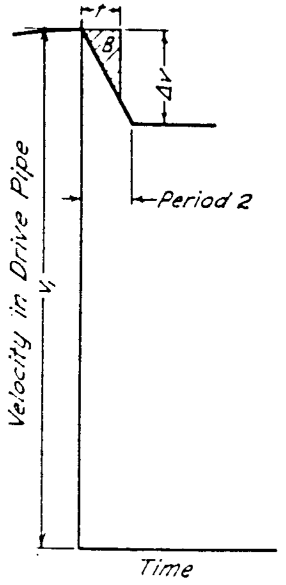


FIG. 4. RELATION OF VELOCITY IN DRIVE PIPE AT RAM TO TIME DURING PERIOD 2

Correspondingly,

$$(\Delta v)_1 = (\Delta v)_2 = \ldots\ldots = \Delta v.$$

where

$$\Delta v = \frac{h_0 g}{a} = \left[(h - H) + \frac{m\,(v_1 - \Delta v)}{4g}\right]\frac{g}{a}. \qquad (22)$$

Such an average introduces no serious error, for when h is high, the friction loss through the check valve is an insignificant part of the pressure variation, and when h is low, there are a large number of surges.

When this assumption is used, Equation (15) becomes

$$v_2 = v_1 - \frac{h_0 g}{a} = v_1 - \Delta v. \qquad (15a)$$

The velocity-time curve for Periods 2 and 3 (Fig. 3), has been drawn according to the assumption of Equation (22) for a cycle having three pressure surges.

The amount of water pumped per cycle is $q_s = \int_0^{t_s} wAv\,dt$, which may be expressed as the product of the specific weight of water, the cross-sectional area of the drive pipe and the area under the velocity-time curve of Fig. 3. The values of the several parts of the area of $\int_0^{t_s} v\,dt$ are indicated on Fig. 3.

The evaluation of areas under the horizontal portions of the velocity-time curve is obvious. To simplify the resulting expressions, the quantity t' is defined as

$$t' = \frac{2L_1}{a} - t_2. \qquad (23)$$

The area under each of the sloping portions of the curve is evaluated in the following manner:

The velocity-time curve for period 2 is shown in Fig. 4. During Period 2, from Equation (14)

$$\frac{dh'}{Y} = Av\,dt.$$

Then $\int \frac{dh'}{YA} = \int v\,dt$ = the area under the velocity-time curve for any part of Period 2. Integration of this equation gives

$$\int_H^{h'} \frac{dh'}{YA} = \frac{h' - H}{YA}.$$

Then the shaded area B in Fig. 4 is

$$B = v_1 t - \frac{h' - H}{YA}.$$

But from Equation (12)

$$(h' - H) = \frac{a}{g}(v_1 - v).$$

Then

$$B = v_1 t - \frac{a}{AgY}(v_1 - v) = v_1 t - Z\,(v_1 - v).$$

From Equation (19),

$$v = v_1 e^{-t/Z}. \qquad (24)$$

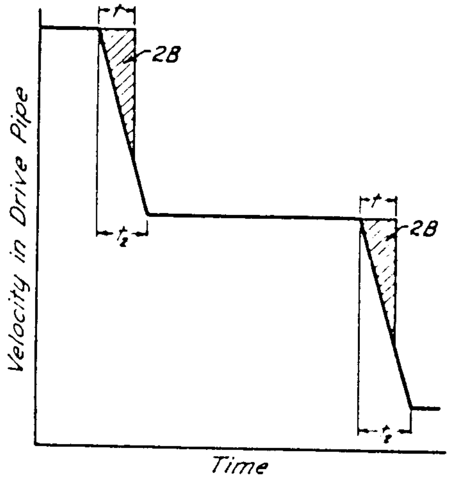


Fig. 5. Relation of Velocity in Drive Pipe at Ram to Time During a Part of Period 3

Hence

$$B = v_1t - v_1Z + v_1Ze^{-t/Z} = v_1\left[t - Z\left(1 - e^{-t/Z}\right)\right]. \quad (25)$$

During Period 2, the decrease in velocity Δv is due to the generation of a pressure wave (Equation (11)); during the sloping portions of the velocity-time curve of Period 3, (see Fig. 5) a pressure wave of identical form is being dissipated and another of equal magnitude is being generated in an equal time interval t_2, reducing the velocity by $2\Delta v$. Therefore the velocity changes which occur at corresponding times during these portions of Period 3 must be twice as great as those during Period 2. Accordingly, the shaded areas of Fig. 5, which is a diagram of the velocity-time curve for part of Period 3, are each twice as great as the shaded area in Fig. 4 for corresponding times. That is, each shaded area of Fig. 5 is equal to

$$2B = 2v_1\left[t - Z\left(1 - e^{-t/Z}\right)\right]. \quad (26)$$

If the shaded area of Fig. 5 be made to include the full width t_2 of a sloping portion of the curve,

$$2B = 2v_1\left[t_2 - Z\left(1 - e^{-t_2/Z}\right)\right]$$

$$2B = 2v_1t_2 - 2Z\left(v_1 - v_1e^{-t_2/Z}\right). \quad (27)$$

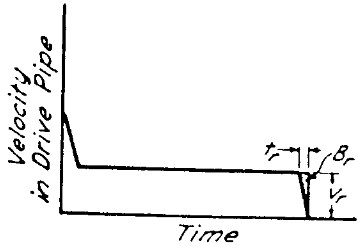


Fig. 6. Relation of Velocity in Drive Pipe at Ram to Time During Final Portion of Period 3

With the aid of Equation (20), Equation (27) is seen to be equivalent to

$$2B = 2v_1t_2 - 2Z\,(v_1 - v_2)$$

or

$$2B = 2v_1t_2 - 2Z\Delta v. \quad (28)$$

Then for the part of surge 2 for which the velocity-time curve is sloping,

$$\int vdt = (v_1 - \Delta v)t_2 - 2B$$
$$= (v_1 - \Delta v)t_2 - (2v_1t_2 - 2Z\Delta v)$$

$$\int vdt = 2Z\Delta v - t_2(v_1 + \Delta v). \quad (29)$$

Similarly, for surge 3, the corresponding area under the sloping part of the velocity-time curve is

$$\int vdt = (v_1 - 3\Delta v)t_2 - 2B$$
$$= (v_1 - 3\Delta v)t_2 - (2v_1t_2 - 2Z\Delta v)$$

$$\int vdt = 2Z\Delta v - t_2(v_1 + 3\Delta v) \quad (30)$$

and so on.

For the last partial surge, the shaded area B_r in Fig. 6, according to Equation (26) is

$$B_r = 2v_1\left[t_r - Z\left(1 - e^{-t_r/Z}\right)\right]$$

$$B_r = 2v_1t_r - 2Z\left(v_1 - v_1e^{-t_r/Z}\right). \quad (31)$$

The term in parentheses in Equation (31) is seen from Equation (24) to be the reduction in velocity during a time t_r of Period 2. Since the velocity decreases twice as fast during t_r as during a corresponding time of Period 2,

$$v_1 - v_1 e^{-t_r/Z} = \frac{v_r}{2}. \tag{32}$$

Substitution in Equation (31) gives

$$B_r = 2v_1 t_r - \frac{2Zv_r}{2}$$

and

$$\int_0^{t_r} v\,dt = v_r t_r - B_r$$

$$= v_r t_r - (2v_1 t_r - Zv_r)$$

$$\int_0^{t_r} v\,dt = Zv_r - t_r\,(2v_1 - v_r). \tag{33}$$

The diagram (Fig. 3) has been drawn for only three surges; if the general consideration of N surges be made, the following equation results:

$$\begin{aligned} q_s = wA\int_0^{t_1} v\,dt = wA\Big\{&\underbrace{[(v_1 - \Delta v)t']}_{\text{surge 1}} \\ &+ \underbrace{[2Z\Delta v - (v_1 + \Delta v)t_2] + [(v_1 - 3\Delta v)t']}_{\text{surge 2}} \\ &+ \underbrace{[2Z\Delta v - (v_1 + 3\Delta v)t_2] + [(v_1 - 5\Delta v)t']}_{\text{surge 3}} \\ &+ \ldots \underbrace{+ Zv_r - (2v_1 - v_r)t_r}_{\text{final partial surge}}\Big\} \\ = wA\,[Nv_1t' &+ (N-1)2Z\Delta v + Zv_r \\ &- N^2\Delta vt' - (N-1)^2\Delta vt_2 - (N-1)v_1t_2 \\ &- (2v_1 - v_r)t_r]. \end{aligned} \tag{34}$$

In Equation (34) Z may be determined from constants of the apparatus, (see Equations (18) and (13)) and v_1 and t_2 are found from Equations (8) and (20). The quantities Δv, N, v_r, and t_r remain to be determined.

From Equation (22)

$$\Delta v = \left[(h - H) + \frac{m(v_1 - \Delta v)}{4g}\right]\frac{g}{a}$$

whence

$$\Delta v = \frac{4g(h - H) + mv_1}{4a + m}. \tag{35}$$

An inspection of Fig. 3 shows that $2\Delta v > v_r \geq 0$, for if v_r were greater than $2\Delta v$, another surge would result, and v_r is prevented from becoming negative by the check valve. But from inspection of Fig. 3, it is seen that

$$v_r = v_1 - (2N - 1)\Delta v. \tag{36}$$

Then

$$2\Delta v > v_1 - 2N\Delta v + \Delta v \geq 0.$$

Addition of $-v_1 - \Delta v$, to the above inequality gives

$$\Delta v - v_1 > -2N\Delta v \geq -v_1 - \Delta v.$$

Division of this result by $-2\Delta v$ gives

$$\frac{v_1 - \Delta v}{2\Delta v} < N \leq \frac{v_1 + \Delta v}{2\Delta v}. \tag{37}$$

It is evident that the difference between the first and last terms of inequality (37) is 1. Since N is necessarily an integer, it is uniquely determined.

With Δv and N determined from Equations (35) and (37), Equation (36) may be used to evaluate v_r.

Since the rate of change of velocity during the time t_r is the same as that during the other sloping portions of Period 3, and is twice as great as during Period 2, at corresponding times,

$$v_r - v\Big]_{\text{Period 3}} = 2(v_1 - v)\Big]_{\text{Period 2}}. \tag{38}$$

Equation (24) states that

$$v\Big]_{\text{Period 2}} = v_1 e^{-t/Z}.$$

Division of both sides by -1, and addition of v_1 to both sides gives

$$v_1 - v\Big]_{\text{Period 2}} = v_1(1 - e^{-t/Z}). \qquad (39)$$

Substitution of Equation (39) in Equation (38) gives

$$v_r - v\Big]_{\text{Period 3}} = 2v_1(1 - e^{-t/Z}).$$

Hence

$$e^{t/Z} = \frac{2v_1}{2v_1 - v_r + v}. \qquad (40)$$

But when $t = t_r$, $v = 0$. Therefore

$$e^{t_r/Z} = \frac{2v_1}{2v_1 - v_r}$$

or

$$t_r = Z \log_e \frac{2v_1}{2v_1 - v_r}. \qquad (41)$$

Since each term in Equation (34) has now been evaluated, q_s may be computed.

By reference to Fig. 3, t_3 is determined as

$$t_3 = \frac{2L_1}{a} N - t_2 + t_r. \qquad (42)$$

If the waste valve is considered inelastic, or $E_v = \infty$, then $Z = 0$, $t_2 = 0$, and Equation (34) reduces to the form given by O'Brien[2] and Gosline,

$$q_s = \frac{2L_1Aw}{a}(Nv_1 - N^2\Delta v). \qquad (43)$$

12. ***Analysis of Period 4.***—At the end of Period 3, the velocity of the water at the check valve is of course zero, and the velocity of the water on the supply side of the pressure wave is $(v_1 - 2N\Delta v)$ which velocity may have any value from $+\Delta v$ (toward the ram) to $-\Delta v$ (away from the ram). The elastic energy stored in that part of the water column between the end of the pressure wave and the check valve is just enough to bring the whole column of water to a final velocity of $(v_1 - 2N\Delta v)$. By definition,

$$(v_1 - 2N\Delta v) = v_3. \qquad (44)$$

There is also some energy stored in the waste valve disc due to its compression. Disregarding energy lost in internal hysteresis and pipe friction, the strain energy in the waste valve disc is expended in giving the water column a velocity toward the supply. The exact pressure and displacement variation is obscure; it may be noted, however, that if v_3 is positive, that is, toward the ram, the velocity must be reversed in direction before the waste valve can open. The reversal is similar to the reversal of velocity caused when a valve is suddenly closed in a straight length of pipe, so that the velocity produced away from the ram is the same in magnitude, but opposite in sense. If v_3 is negative, no such reversal is required.

The energy stored in water velocity and in compression of the waste valve disc is evaluated as follows:

$$\text{Energy stored in waste valve disc} = \tfrac{1}{2}(\text{load}) \times (\text{deflection}). \qquad (45)$$

However,

$$\text{load} = wh_0A_v$$

and

$$\text{deflection} = \frac{\text{load}}{E_v} = \frac{wh_0A_v}{E_v}$$

Then

$$\text{Energy stored in waste valve disc} = \tfrac{1}{2}\, wh_0A_v \frac{wh_0A_v}{E_v}. \qquad (46)$$

From Equation (22)

$$h_0 = \frac{\Delta va}{g}$$

$$\text{Energy stored in waste valve disc} = \tfrac{1}{2}\frac{w^2A_v{}^2a^2\Delta v^2}{g^2E_v}. \qquad (47)$$

But since

$$Z = \frac{a}{AgY} = \frac{awA_v^2}{AgE_v}$$

$$\text{Energy stored in waste valve disc} = \frac{wA}{2g} aZ(\Delta v)^2 \qquad (48)$$

$$\text{Energy due to velocity of water column} = \tfrac{1}{2}(\text{mass})$$

$$\times (\text{velocity})^2 = \tfrac{1}{2}\frac{wAL}{g}v^2. \qquad (49)$$

The kinetic energy due to the velocity v_3, added to the strain energy stored in the waste valve disc, is assumed equal to the kinetic energy required to produce the velocity v_4. That is,

$$\frac{wA}{2g} Lv_3^2 + \frac{wA}{2g} aZ(\Delta v)^2 = \frac{wA}{2g} Lv_4^2$$

whence

$$v_4 = -\sqrt{v_3^2 + \frac{aZ}{L}(\Delta v)^2}\,. \qquad (50)$$

The time during which these energy transfers take place is not determined by the energy analysis, since the actual velocity variations are obscure. But it seems reasonable to assume that if v_3 is positive, so that a reversal takes place, this reversal will take a time $2L_1/a$. Therefore if v_3 is positive, t_4 will be $2L_1/a$ seconds longer than if v_3 is negative.

If v_3 is negative, it is evident that enough water must flow out of the valve box during Period 4 to make room for the expansion of the waste valve to its uncompressed size. Therefore,

$$\text{flow out of the valve box} = \text{volume of expansion of waste valve disc.} \qquad (51)$$

The volume of expansion of the waste valve disc must equal the volume of compression during Period 2. But in Period 2, since the volume of water flowing into the valve box must equal the volume of compression,

$$\text{volume of compression} = A\int_0^{t_2} v\,dt$$

for Period 2.

From Fig. 4,

$$\int_0^{t_2} v\,dt = v_1t_2 - B$$

for Period 2. Evaluation of B from Equation (25) when $t = t_2$ gives

$$A\int_0^{t_2} v\,dt = A\left[v_1t_2 - v_1\left\{t_2 - Z(1 - e^{-t_2/Z})\right\}\right] = AZ(v_1 - v_1e^{-t_2/Z})$$

or

$$\text{volume of compression} = AZ(v_1 - v_1e^{-t_2/Z}).$$

By Equation (24)

$$(v_1 - v_1e^{-t_2/Z}) = v_1 - v_2 = \Delta v.$$

Hence

$$\text{volume of compression} = AZ\Delta v.$$

The total volume which flows out of the valve box during Period 4 must equal the product of the average velocity, the area of the pipe, and the time of the period. An approximate average velocity is $\frac{v_3 + v_4}{2}$.

Therefore,

$$\text{flow out of the valve box} = \frac{v_3 + v_4}{2} At_4.$$

Substitution of these values in Equation (51) gives

$$-\frac{v_3 + v_4}{2} At_4 = AZ\Delta v.$$

The negative sign on the left-hand side is necessary because velocity has been defined as negative when directed away from the ram. Thus, if v_3 is negative,

$$t_4 = -\frac{2Z\Delta v}{(v_3 + v_4)}. \qquad (52a)$$

If v_3 is positive,

$$t_4 = -\frac{2Z\Delta v}{(v_4 - v_3)} + \frac{2L_1}{a}. \qquad (52b)$$

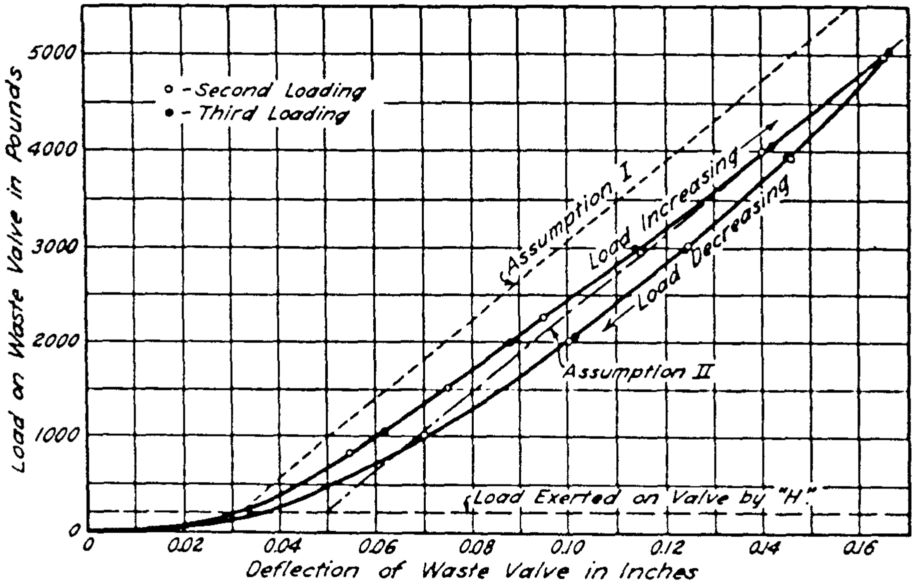


FIG. 7. LOAD-DEFLECTION TEST OF WASTE VALVE OF THE 4-INCH RAM

By Equations (50) and (52a) or (52b) the final velocity v_4 (negative) and the length (t_4) of Period 4 may be determined.

The preceding analysis has all been made on the assumption that the load-deflection curve for the waste valve is the straight line of assumption I of Fig. 7. Another assumption which might have been made is also shown in Fig. 7. Assumption II is probably closer to the actual curve, but introduces some inconvenience in analysis. Enough study of assumption II has been made to show the following differences in the two assumptions.

During Period 2, both assumptions give the same results, since the equations in Period 2 require only that the curve be a straight line with slope E_r, and both assumptions fulfill this requirement. Thus the value of t_2 is the same for both assumptions, and therefore the results in Period 3 are unaffected by the choice of assumption.

During Period 4, the energy analysis which gives the relation from which v_4 is calculated (Equation (50)) is the same for both assumptions, since the energy for any load on the valve is equal to the area under the load-deflection curve, and that area is the same for both assumptions.

The displacement of the water which must take place in order that the waste valve can open at the end of Period 4 is proportional to the actual displacement of the valve. The two assumptions give considerably different results for this quantity, as shown in Fig. 7. However, the time required to accomplish the displacement which assumption II would involve is never more than 0.005 seconds greater than the time (t_4) which assumption I requires for the 2-inch ram. For the 4-inch ram the difference is greater; for the worst case of very low pumping head and very low v_0, the difference in the duration of Period 4 as determined according to the two assumptions is 0.042 seconds, or 6 per cent of the total time t_c. For the condition of maximum efficiency for any v_0, the difference is never greater than 3 per cent of the total time t_c. The difference decreases rapidly with the increase of either pumping head or v_0.

Of course, during Periods 1, 5, and 6 the assumption used makes no difference, since no deflection of the waste valve occurs.

The convenience of assumption I seems to justify its use, since the errors produced by using it have been shown to be comparatively small.

13. *Analysis of Period 5.*—During this period the water is considered incompressible. Pressure fluctuations during Periods 5 and 6 are small enough to make the effects of compressibility negligible. The velocity in the drive pipe is therefore uniform all along the pipe, and is at first negative, equal to v_4. The water is continually accelerated toward the ram by the unbalanced supply pressure, so that the velocity decreases to zero, then increases again toward the ram, until the valve box is refilled and water begins to waste.

Pipe friction is disregarded during this period. An analysis was made taking friction into account, and numerical calculations showed that for the test ranges of the rams used experimentally, the values of v_5 and t_5, as determined by disregarding friction, were not more than 5 per cent greater than the values determined by taking friction into account.

Consideration of the forces on the water column during Period 5 gives

$$\text{Force} = \text{mass} \times \text{acceleration}$$

$$wAH = \frac{wAl}{g} \times \frac{dv}{dt}$$

where l is the variable length of the water column in the drive pipe.

However, the variation in l is quite small compared to the maximum length L. The substitution $l = L$ gives

$$dt = \frac{L}{gH}\,dv.$$

$$\int_0^{t_5} dt = \int_{v_4}^{v_5} \frac{L}{gH}\,dv.$$

Therefore

$$t_5 = \frac{L}{gH}(v_5 - v_4).$$

Since, if friction is disregarded,

$$v_5 = -v_4 \qquad (53)$$

$$t_5 = \frac{-2Lv_4}{gH} = \frac{2Lv_5}{gH}. \qquad (54)$$

From Equations (53) and (54) respectively, v_5 and t_5 may be determined.

14. *Analysis of Period 6.*—Friction must be taken into account during this period, since there is appreciable loss both in drive pipe and waste valve, with higher velocities.

Consideration of the forces acting on the column of water in the drive pipe gives

Force = mass × acceleration

$$wA\left(H - \frac{v^2}{2g}\right) - wA(b + c)\frac{v^2}{2g} = wA\,\frac{L}{g}\,\frac{dv}{dt}.$$

The substitution $j = b + c + 1$ gives

$$dt = \frac{\dfrac{2L}{j}}{\dfrac{2gH}{j} - v^2}\,dv \qquad (55)$$

$$\int_0^{t_6} dt = \int_{v_5}^{v_6} \frac{\dfrac{2L}{j}\,dv}{\dfrac{2gH}{j} - v^2}.$$

Therefore

$$t_6 = \frac{L}{j\sqrt{\dfrac{2gH}{j}}}\log_e \frac{\dfrac{\sqrt{\dfrac{2gH}{j}} + v_6}{\sqrt{\dfrac{2gH}{j}} - v_6}}{\dfrac{\sqrt{\dfrac{2gH}{j}} + v_5}{\sqrt{\dfrac{2gH}{j}} - v_5}}.$$

Substitution of v_0 for v_6 gives

$$t_6 = \frac{L}{j\sqrt{\dfrac{2gH}{j}}}\log_e \frac{\left(\sqrt{\dfrac{2gH}{j}} + v_0\right)}{\left(\sqrt{\dfrac{2gH}{j}} - v_0\right)}\,\frac{\left(\sqrt{\dfrac{2gH}{j}} - v_5\right)}{\left(\sqrt{\dfrac{2gH}{j}} + v_5\right)}. \qquad (56)$$

From Equation (56) t_6 may be determined.

From Equation (55) it is seen that

$$\frac{dv}{dt} = \frac{\dfrac{2gH}{j} - v^2}{\dfrac{2L}{j}} \text{ at any time during Period 6.}$$

Since α_6 is defined as the acceleration of the water column at the end of Period 6 when the velocity is v_6, or its equivalent v_0,

$$\alpha_6 = \frac{\dfrac{2gH}{j} - v_0^2}{\dfrac{2L}{j}}. \qquad (57)$$

TABLE 1
OUTLINE OF METHOD OF COMPUTATION
Beginning at Period 1

Period	Velocity-Functions	Equation No.	Time-Functions	Equation No.	Displacement-Functions	Equation No.	Constants to be Determined
1	$v_1 = f(p, t_1)$	8	$t_1 = f(p)$	4	$Q_1 = f(p, t_1)$	10	S_0, J, w b, c, v_0, g, H, L, A
2	$v = f(p, v_1)$	15	$t_2 = f(p, v_1, v_2)$	20			L_1, h, m, A_s, E_s, a
3	$v_2 = f(p, v_1, v_2)$	44	$t_3 = f(p, v_1, v_2, t_2)$	42	$q_e = f(p, v_1, v_2, t_2)$	34	
4	$v_4 = f(p, v_1, v_2, v_3)$	50	$t_4 = f(p, v_1, v_2, v_3, v_4)$	52			
5	$v_5 = -v_4$	53	$t_5 = f(p, v_4)$	54			
6	$v_6 = v_0$		$t_6 = f(p, v_5)$	56	$Q_6 = f(p, v_5)$	58	

Also from the equation for dv/dt, the quantity wasted during Period 6 is

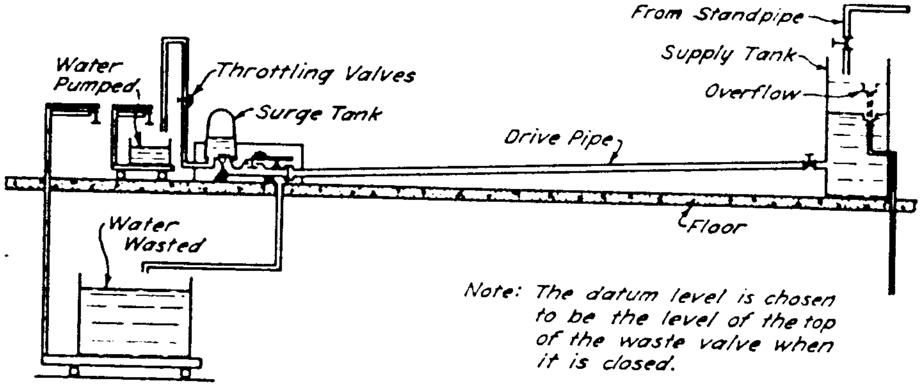
$$Q_6 = Aw\int vdt = Aw\int_{v_5}^{v_6} \frac{\frac{2L}{j}vdv}{\frac{2gH}{j} - v^2}.$$

Therefore

$$Q_6 = \frac{wAL}{j}\log_e \frac{\frac{2gH}{j} - v_5^2}{\frac{2gH}{j} - v_0^2}. \qquad .(58)$$

15. *Summary of Analysis.*—Inspection of all the equations so far developed will show that though they have been worked out assuming a horizontal drive pipe, they are independent of the slope of the drive pipe as long as all parts of the drive pipe are under a sufficient absolute pressure to prevent cavitation.

To show the continuity of the analysis, and the sequence in which computations must be made, Table 1 is presented. The symbol p is understood to be indicative of constants of the setting of the ram which can be measured or assumed.



Note: The datum level is chosen to be the level of the top of the waste valve when it is closed.

FIG. 8. DIAGRAM OF LABORATORY SETTING

Now that the analysis has been completed in each period, the results are combined to obtain the desired quantities Q and q.

$$t_e = t_1 + t_2 + t_3 + t_4 + t_5 + t_6 \qquad (59)$$

$$Q_e = Q_1 + Q_6 \qquad (60)$$

$$Q = \frac{60Q_e}{t_e} \qquad (61)$$

$$q = \frac{60q_e}{t_e}. \qquad (62)$$

III. EXPERIMENTAL STUDY

16. *Pumping Tests.*—Two rams were used to obtain experimental performance data. Both were manufactured by the Rife Hydraulic Manufacturing Company; one which was designated by them as size No. 40A was fitted with a 4-in. drive pipe 55.5 ft. long; the other, which was size No. 20B, was fitted with a 2-in. drive pipe 54.8 ft. long. Figure 8 is a diagram of the laboratory setting which was similar for the two rams. Figures 9 and 10 are views of the 4-in. ram as set up for operation.

Water was supplied from a constant-head tank, equipped with two overflow pipes at different levels so that two supply pressures were available. Only the higher of these, approximately nine feet,

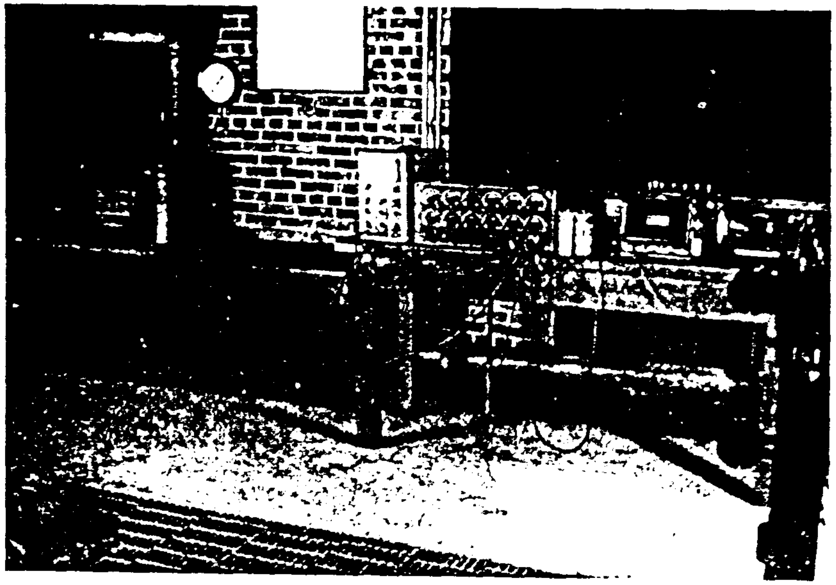


FIG. 9. VIEW OF RAM AND OSCILLOGRAPH AS SET UP IN THE LABORATORY

was used in the performance tests, since an unsteady operation was observed when operation at the five-foot supply head was attempted. The supply pressure was measured by a manometer fitted with a gage reading to 0.01 ft. Though some surges occurred in the manometer due to the pounding of the ram and the shaking of the supply tank, the average supply pressure did not vary more than 0.01 ft. during any test.

The delivery pressure was controlled by throttling valves in the delivery pipe. This pressure was measured by a Bourdon gage of suitable range, calibrated by a dead-weight tester before and after use. The readings of each gage were corrected for errors in indication and for the height of the gage above datum. Particularly at high delivery pressures, some difficulty was experienced in maintaining a constant discharge pressure, and some adjustment of the valves was found necessary during the progress of a test. The pressure was allowed to vary no more than one per cent on either side of the mean indicated pressure.

The water wasted during operation of the ram was caught in a box which surrounded the ram, and drained to a tank mounted on scales. The rate of wasting was determined by measuring, with an Eastman timer reading to half-seconds, the time required for wasting a given quantity of water. Both timer and scales were calibrated,

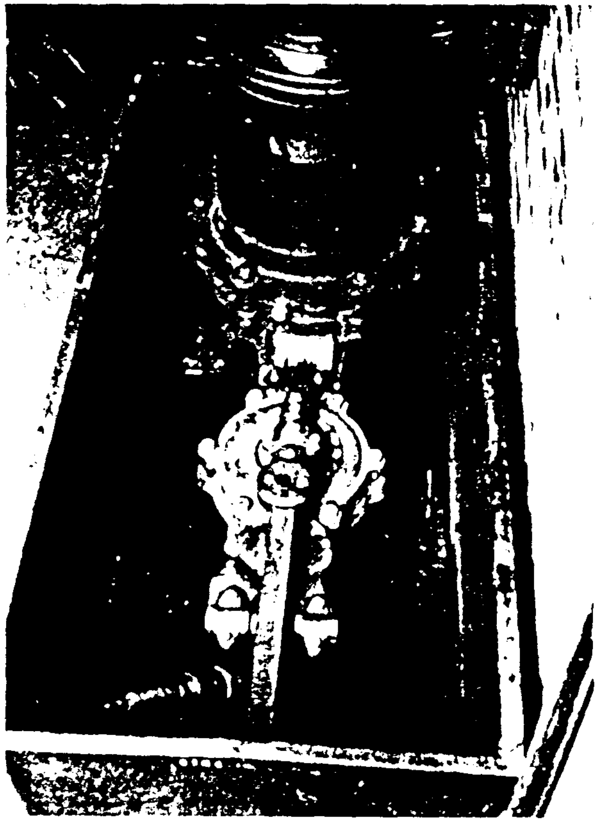


FIG. 10. VIEW OF RAM AND ELECTRIC PRESSURE GAGE AS SET UP IN THE LABORATORY

and no test of less than two minutes was made. The temperature of the water was approximately 70 deg. F.

The water pumped was measured in another tank, mounted on calibrated scales. The rate of pumping was determined by measuring with another calibrated timer the time required for pumping a given quantity of water. No test of less than $2\frac{1}{2}$ minutes was made, except at extremely low delivery pressures, when the capacity of the tank limited the length of the test.

The frequency of the pumping cycle was determined by measuring, with one of the previously described timers, the time for 100 cycles. For all the tests the frequency was less than 100 cycles per minute.

The velocity, v_o, in the drive pipe necessary to begin waste valve closure, was controlled by two adjustments. The length of the

TABLE 2
PERFORMANCE DATA

Series 1

2-in. ram; $H = 9.2$ ft.; $v_0 = 4.48$ ft. per sec.; $S_0 = 0.0269$ ft.; weight to the left

Test Number	h ft.	Q lb. per min.	q lb. per min.	Rankine Efficiency* per cent	Q_e lb. per cycle	q_e lb. per cycle	t_e sec. per cycle
711	24.5	176.0	30.6	28.9	4.40	0.765	1.450
710	34.0	182.2	25.0	30.9	4.38	0.600	1.440
709	43.5	184.9	21.0	42.4	4.38	0.498	1.420
708	53.0	185.2	17.9	46.0	4.36	0.421	1.415
707	63.5	187.5	15.7	49.4	4.34	0.364	1.390
706	73.5	187.5	13.8	51.5	4.33	0.319	1.385
705	83.5	187.5	12.2	52.6	4.33	0.282	1.385
704	93.5	184.6	10.9	54.1	4.30	0.254	1.395
703	103.5	185.2	9.9	54.7	4.31	0.231	1.395
702	113.5	185.2	9.1	55.7	4.29	0.211	1.390
701	123.5	180.7	8.0	55.0	4.25	0.188	1.415
700	133.0	177.5	7.3	55.1	4.23	0.174	1.430
699	139	179.7	7.1	55.6	4.28	0.169	1.430
698	149	179.1	6.5	55.3	4.27	0.155	1.430
697	159	180.2	6.1	55.2	4.25	0.143	1.415
671	168	182.3	5.9	55.9	4.33	0.140	1.420
672	177	181.8	5.5	55.4	4.30	0.130	1.420
673	188	176.4	4.9	54.1	4.25	0.118	1.445
674	197	172.0	4.6	54.5	4.21	0.112	1.465
675	206	164.9	4.1	52.7	4.09	0.102	1.490
676	216	159.0	3.8	53.8	4.00	0.096	1.510
677	226	158.5	3.6	54.0	4.00	0.091	1.515
678	235	157.7	3.5	54.6	3.98	0.088	1.515
679	246	157.7	3.3	54.0	3.97	0.083	1.510
680	256	158.8	3.2	54.2	3.98	0.080	1.505
681	266	159.0	3.1	54.6	3.99	0.078	1.505
682	275	158.8	2.9	53.0	3.98	0.073	1.505
683	286	159.7	2.8	53.3	3.99	0.070	1.500
684	297	160.4	2.7	52.7	4.01	0.068	1.500
685	307	159.8	2.6	52.7	4.00	0.065	1.500
686	317	161.3	2.5	54.0	4.01	0.062	1.495
687	327	161.3	2.3	49.3	4.01	0.057	1.495
688	337	161.3	2.2	48.0	4.01	0.055	1.495
689	346	160.8	2.1	47.8	3.99	0.052	1.490
690	357	160.4	1.9	44.7	3.99	0.047	1.495
691	367	157.4	1.8	44.5	3.94	0.045	1.500
692	378	154.1	1.6	41.7	3.89	0.040	1.510
693	387	151.0	1.5	40.8	3.82	0.039	1.515
694	397	148.3	1.4	39.9	3.78	0.036	1.530
695	407	144.9	1.2	35.8	3.74	0.031	1.545
696	418	142.1	1.1	34.4	3.68	0.029	1.555

Series 2

2-in. ram; $H = 9.2$ ft.; $v_0 = 2.93$ ft. per sec.; $S_0 = 0.0269$ ft.; weight to the right

Test Number	h ft.	Q lb. per min.	q lb. per min.	Rankine Efficiency* per cent	Q_e lb. per cycle	q_e lb. per cycle	t_e sec. per cycle
712	29.0	100.0	22.4	48.1	1.53	0.343	0.920
713	34.0	102.3	20.2	53.2	1.54	0.305	0.905
714	43.5	105.5	16.7	60.8	1.55	0.216	0.885
715	53.0	105.8	13.8	62.1	1.55	0.202	0.880
716	63.5	104.4	11.5	65.0	1.53	0.169	0.885
717	73.5	105.3	10.1	67.0	1.54	0.147	0.875

*Rankine efficiency $= \dfrac{q(h - H) \times 100}{QH}$.

TABLE 2 (CONTINUED)
PERFORMANCE DATA

Series 2 (Concluded)

2-in. ram; $H = 9.2$ ft.; $v_0 = 2.93$ ft. per sec.; $S_0 = 0.0269$ ft.; weight to the right

Test Number	h ft.	Q lb. per min.	q lb. per min.	Rankine Efficiency* per cent	Q_e lb. per cycle	q_e lb. per cycle	t_e sec. per cycle
718	83.5	102.1	8.5	67.2	1.52	0.127	0.895
719	93.5	100.5	7.5	68.3	1.50	0.112	0.900
720	103.5	100.5	6.8	69.4	1.50	0.102	0.900
721	113.5	101.8	6.2	69.2	1.53	0.093	0.900
722	123.5	99.9	5.5	68.5	1.50	0.082	0.900
723	133.0	94.8	4.7	66.7	1.45	0.072	0.920
724	139	92.0	4.4	66.6	1.44	0.068	0.930
725	150	87.7	3.7	65.0	1.39	0.059	0.950
726	160	87.6	3.5	66.0	1.39	0.056	0.955
727	169	87.0	3.3	66.3	1.39	0.053	0.955
728	178	86.5	3.1	65.8	1.37	0.049	0.950
729	188	85.6	2.8	65.0	1.35	0.045	0.950
730	197	86.5	2.6	63.0	1.36	0.042	0.945
731	207	87.0	2.5	61.2	1.37	0.039	0.945
732	216	87.6	2.3	59.2	1.37	0.036	0.940
733	226	87.4	2.1	56.8	1.37	0.033	0.940
734	235	86.8	1.9	53.8	1.37	0.030	0.950
735	246	85.2	1.7	51.2	1.36	0.027	0.955
736	256	82.8	1.5	48.5	1.33	0.024	0.960
737	266	80.0	1.3	45.3	1.30	0.021	0.975
738	276	78.5	1.2	44.6	1.30	0.020	0.990
739	285	75.6	1.0	39.8	1.28	0.017	1.015
740	296	73.8	0.9	36.8	1.27	0.015	1.035
741	307	69.3	0.7	34.8	1.21	0.013	1.045
742	317	63.8	0.5	26.9	1.12	0.009	1.055
743	327	58.8	0.3	19.8	1.05	0.006	1.075
744	337	53.6	0.2	11.0	0.97	0.003	1.085

Series 3

2-in. ram; $H = 9.2$ ft.; $v_0 = 1.77$ ft. per sec.; $S_0 = 0.0108$ ft.; weight to center.

Test Number	h ft.	Q lb. per min.	q lb. per min.	Rankine Efficiency* per cent	Q_e lb. per cycle	q_e lb. per cycle	t_e sec. per cycle
745	24.5	54.6	16.1	48.7	0.62	0.182	0.680
746	34.0	57.8	12.3	56.8	0.63	0.133	0.650
747	43.5	58.8	9.6	61.2	0.61	0.100	0.625
748	53.0	58.3	7.4	60.3	0.60	0.076	0.615
749	63.5	58.5	6.2	62.9	0.60	0.064	0.610
750	73.5	55.2	5.2	66.2	0.57	0.054	0.615
751	83.5	54.4	4.2	63.8	0.57	0.045	0.625
752	93.5	53.5	3.4	57.0	0.57	0.036	0.635
753	103.5	50.4	2.9	58.8	0.54	0.031	0.640
754	113.5	49.0	2.6	59.8	0.53	0.028	0.655
755	123.5	48.2	2.3	60.9	0.53	0.026	0.665
756	133.0	48.1	2.1	58.5	0.53	0.023	0.665
757	139	48.5	2.0	58.5	0.53	0.022	0.660
758	150	48.3	1.7	54.7	0.53	0.019	0.660
759	160	46.8	1.4	50.4	0.52	0.016	0.670
760	170	44.9	1.1	45.5	0.50	0.013	0.670
761	179	44.0	0.9	40.6	0.50	0.011	0.685
762	189	41.3	0.7	32.5	0.48	0.008	0.700
763	198	38.0	0.4	22.9	0.45	0.005	0.705
764	207	34.9	0.2	10.3	0.42	0.002	0.725

*Rankine efficiency $= \dfrac{q(h - H) \times 100}{QH}$.

Table 2 (Continued)
Performance Data
Series 4

4-in. ram; H = 9.0 ft.; v_0 = 3.19 ft. per sec.; S_0 = 0.0301 ft.; weight to the left

Test Number	h ft.	Q lb. per min.	q lb. per min.	Rankine Efficiency* per cent	Q_e lb. per cycle	q_e lb. per cycle	t_e sec. per cycle
213	23.5	409	146.2	57.6	8.53	3.05	1.250
214	33.5	438	106.5	66.1	8.40	2.04	1.150
215	43.0	449	81.3	68.5	8.28	1.50	1.105
216	52.5	455	64.8	69.0	8.35	1.19	1.100
217	63.0	452	52.0	69.1	8.22	0.946	1.090
218	73.0	453	44.5	69.8	8.27	0.812	1.095
219	83.0	445	37.9	70.0	8.13	0.692	1.095
220	93.0	433	32.0	69.0	7.95	0.588	1.100
221	103.0	422	27.2	67.4	7.95	0.513	1.130
222	113.0	412	24.4	68.3	7.81	0.462	1.135
223	122.5	404	21.6	67.4	7.68	0.411	1.140
224	132.5	395	19.1	66.2	7.57	0.366	1.150
225	139	388	17.8	66.5	7.51	0.345	1.160
226	150	374	15.1	63.2	7.34	0.296	1.175
227	160	356	12.5	58.9	7.12	0.250	1.200
228	170	348	11.3	57.8	7.03	0.227	1.210
229	179	335	9.7	55.1	6.78	0.198	1.215
230	189	328	8.9	54.1	6.67	0.181	1.220
231	198	322	8.1	52.9	6.55	0.165	1.220
232	207	317	7.5	51.8	6.53	0.154	1.235
233	217	311	6.8	50.0	6.46	0.140	1.245
234	227	304	6.0	47.1	6.31	0.123	1.245
235	236	297	5.2	44.1	6.16	0.108	1.245
236	247	286	4.4	40.2	5.99	0.091	1.255
237	258	279	3.9	38.5	5.90	0.082	1.270
238	267	271	3.4	36.1	5.80	0.073	1.285
239	276	270	2.9	32.1	5.83	0.063	1.295
240	287	256	2.1	25.5	5.57	0.046	1.305
241	298	243	1.5	19.8	5.33	0.033	1.315
242	309	234	1.0	14.8	5.19	0.023	1.330
243	318	225	0.6	9.6	5.00	0.014	1.335
244	328	219	0.2	3.6	4.91	0.005	1.345

Series 5

4-in. ram; H = 9.0 ft.; v_0 = 2.25 ft. per sec.; S_0 = 0.0204 ft.; weight to the left

Test Number	h ft.	Q lb. per min.	q lb. per min.	Rankine Efficiency* per cent	Q_e lb. per cycle	q_e lb. per cycle	t_e sec. per cycle
258	24.0	267	113.3	71.0	4.20	1.786	0.945
259	33.5	291	79.0	73.8	4.17	1.131	0.860
260	43.0	296	58.8	75.1	4.12	0.818	0.835
261	53.0	297	45.5	74.8	4.06	0.621	0.820
262	62.5	290	36.4	74.6	3.98	0.499	0.825
263	72.5	282	29.5	73.9	3.92	0.410	0.835
264	82.5	267	23.5	71.7	3.77	0.331	0.845
265	93.0	262	20.0	71.5	3.77	0.289	0.865
266	103.0	256	17.0	69.4	3.73	0.248	0.875
267	113.0	250	14.2	66.6	3.75	0.213	0.900
268	123.0	240	11.6	61.0	3.68	0.177	0.920
269	132.5	231	9.8	58.3	3.60	0.153	0.935
270	138	230	9.1	56.8	3.60	0.143	0.940
271	149	220	7.8	54.8	3.50	0.123	0.945
272	159	216	6.8	52.8	3.45	0.109	0.960
273	168	210	5.9	50.1	3.39	0.096	0.970
274	178	203	5.1	48.8	3.32	0.083	0.980
275	188	195	4.0	41.1	3.20	0.066	0.985
276	197	187	3.3	36.5	3.09	0.054	0.990
277	206	177	2.5	30.7	2.93	0.041	0.995
278	216	172	1.8	23.8	2.90	0.030	1.010
279	226	165	1.1	14.6	2.78	0.018	1.010
280	236	157	0.7	11.4	2.66	0.012	1.015
281	246	151	0.2	3.0	2.59	0.003	1.030

*Rankine efficiency = $\frac{q(h - H) \times 100}{QH}$.

Table 2 (Concluded)
Performance Data
Series 6

4-in. ram; H = 9.0 ft.; v_0 = 1.35 ft. per sec.; S_0 = 0.0164 ft.; weight to the right

Test Number	h ft.	Q lb. per min.	q lb. per min.	Rankine Efficiency* per cent	Q_e lb. per cycle	q_e lb. per cycle	t_e sec. per cycle
245	24.5	174	72.0	71.2	1.96	0.810	0.675
246	34.5	184	48.4	74.6	1.93	0.508	0.630
247	44.5	184	34.5	74.1	1.87	0.351	0.610
248	54.5	175	25.6	73.9	1.78	0.260	0.610
249	64.0	160	18.8	71.9	1.65	0.192	0.615
250	74.5	150	14.2	68.9	1.58	0.149	0.630
251	85.0	142	11.1	66.0	1.53	0.119	0.645
252	95.5	131	8.0	58.8	1.41	0.089	0.665
253	107.0	126	6.2	53.7	1.41	0.069	0.675
254	110	121	5.4	50.3	1.36	0.061	0.675
255	118	115	4.3	45.3	1.31	0.049	0.685
256	127	104	2.9	35.8	1.21	0.033	0.695
257	138	93	1.5	25.0	1.09	0.019	0.700

*Rankine efficiency = $\frac{q(h - H) \times 100}{QH}$.

stroke of the waste valve was adjustable by means of two screws (L in Fig. 1). The counterweight (M in Fig. 1) was adjustable along the rocker arm of the waste valve. Both of these adjustments affect v_0. For each ram, a series of tests was made with each of three different values of v_0. In each series the delivery pressure was varied in approximately equal increments over the whole range of the ram, all other factors being held constant.

The results of these series of tests are presented in Table 2, and are shown graphically in Figs. 11 to 14 inclusive.

17. *Determination of Constants.*—The following constants were required (see Table 1):

$$S_0, J, m, b, c, v_0, E_v, L, L_1, A, A_v, a.$$

S_0, the length of stroke of the waste valve, was determined by attaching a steel rod to the center of the waste valve disc, so that it extended up to an Ames dial graduated in thousandths of an inch, mounted on the box surrounding the ram. Since it was difficult to determine the exact point at which the waste valve first touched its seat, and since the valve did not strike its seat at all points at once, S_0 was arbitrarily defined as the distance between the open position of the waste valve and its position when held closed by a pressure head of approximately nine feet of water. This value was easily determined with the Ames dial.

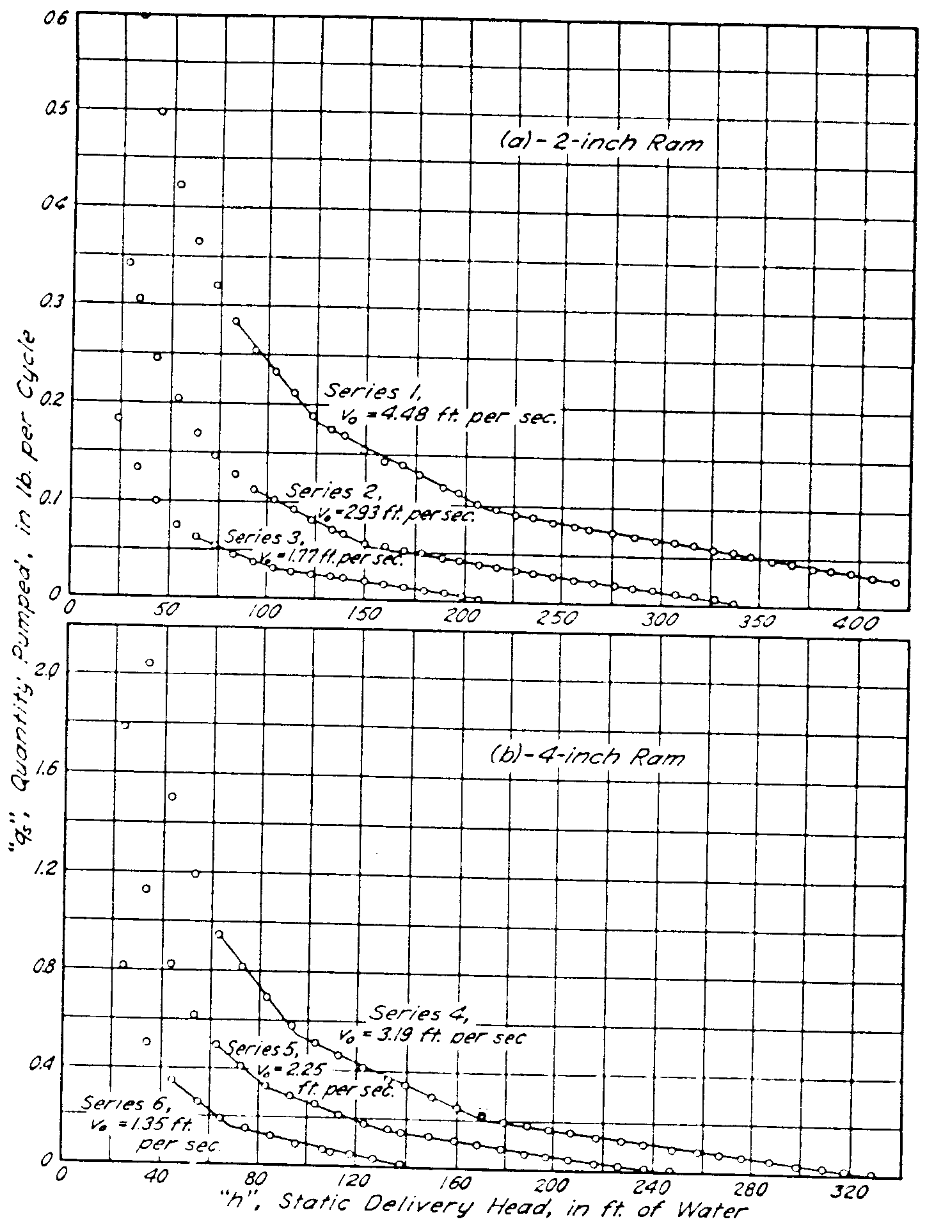


FIG. 11. RELATION OF q_s, THE QUANTITY PUMPED PER CYCLE, TO h, THE STATIC DELIVERY HEAD, AS DETERMINED BY EXPERIMENT FOR THE 2-INCH RAM AND THE 4-INCH RAM

A brass stylus was attached to the upper end of the steel rod used in the determination of S_0 in such a manner that it could be brought to bear on a strip of sensitized paper mounted on a drum which revolved at constant speed. Records of the relation of waste valve movement to time were thus obtained for a number of conditions of operation. Four of these records are reproduced in Fig. 15. The results of measurements of the length of time of waste valve closure from more than eighty of these records from the 4-in. ram are summarized in Table 3. These results apparently justified the assumption that t_1 could be represented by a function of $\sqrt{S_0}$

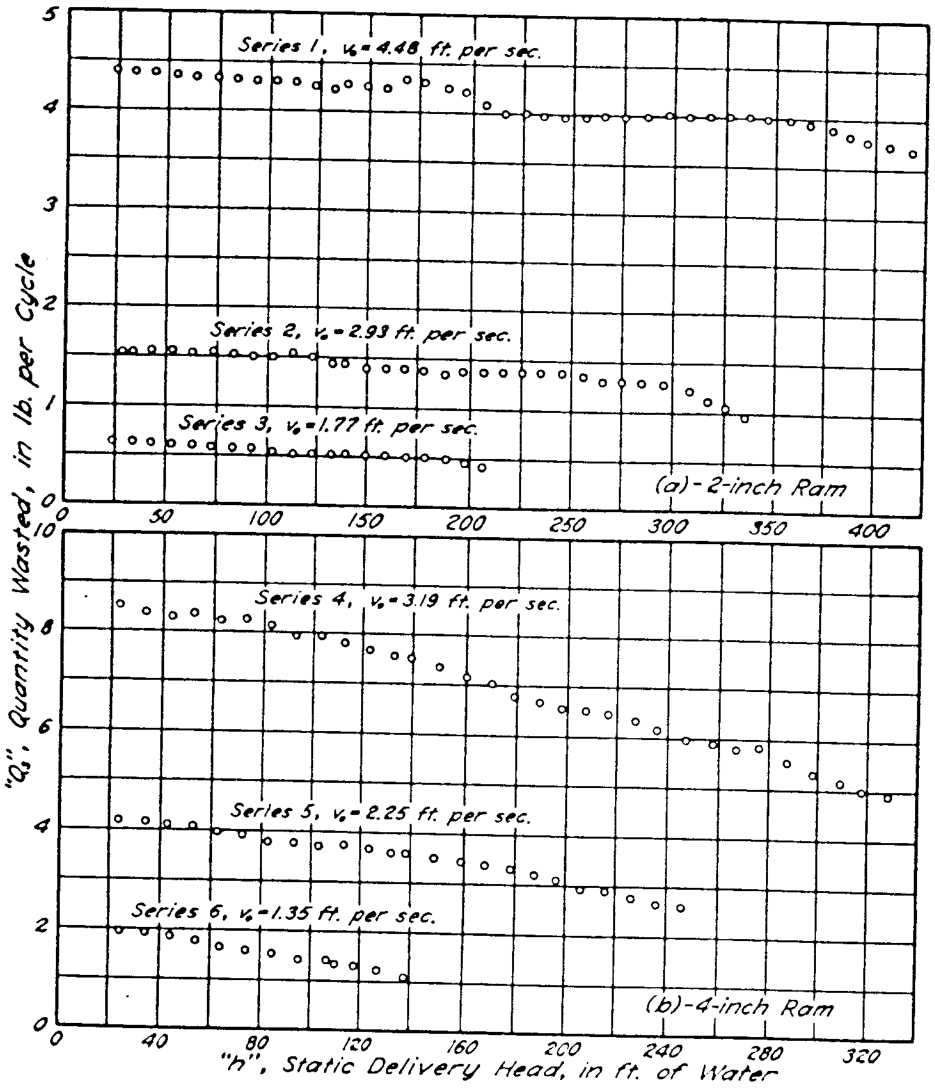


FIG. 12. RELATION OF Q_s, THE QUANTITY WASTED PER CYCLE, TO h, THE STATIC DELIVERY HEAD, AS DETERMINED BY EXPERIMENT FOR THE 2-INCH RAM AND THE 4-INCH RAM

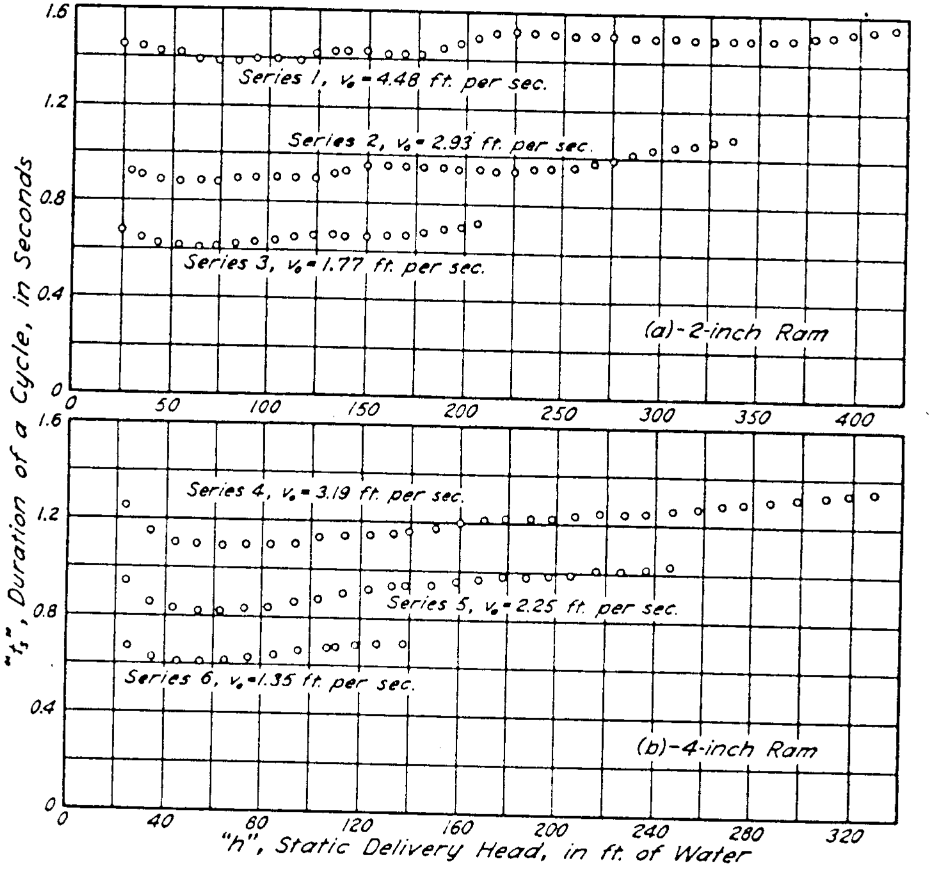


FIG. 13. RELATION OF t_1, THE DURATION OF A CYCLE, TO h, THE STATIC DELIVERY HEAD, AS DETERMINED BY EXPERIMENT FOR THE 2-INCH RAM AND THE 4-INCH RAM

alone, without reference to the counterweight position or supply or delivery pressure (Equation (4)). This assumption established the form of the function. The constant J was then evaluated for each ram by using the amounts actually wasted as a criterion of how long the closure period should be. In Table 4 are compiled a group of measurement results from the displacement–time diagrams for the waste valve, and the corresponding calculated values; it is evident that the discrepancies are considerable. The difficulty of precise determination of the point on the autographic records where the waste valve begins to move is probably the cause of part of these discrepancies.

The friction constants m, b, and c were determined by steady-flow tests, head loss being measured by a water manometer. To

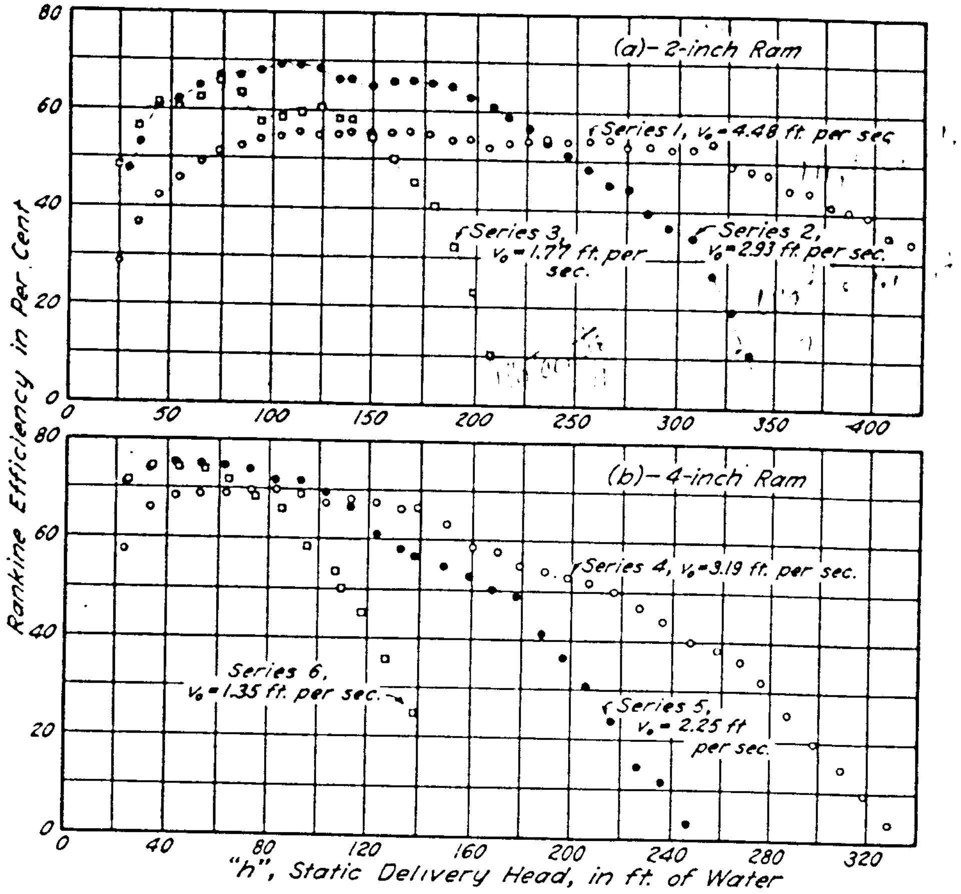


FIG. 14. RELATION OF RANKINE EFFICIENCY TO h, THE STATIC DELIVERY HEAD, AS DETERMINED BY EXPERIMENT FOR THE 2-INCH RAM AND THE 4-INCH RAM

determine m, the check valve constant, the dome of the surge tank was removed, and the flow controlled by throttling with the gate valve at the upper end of the drive pipe. The values of head loss were then plotted as ordinates and the corresponding values of velocity as abscissas in rectangular coordinates, and the slope of the straight line which best fitted the plotted points was multiplied by $2g$ to obtain m. To determine b, the friction coefficient of the drive pipe, the flow was controlled by blocking the waste valve partly open. To determine c, the friction coefficient of the waste valve, the flow was controlled by throttling at the upper end of the drive pipe, and the velocity always kept below that value v_0 which would cause the waste valve to close.

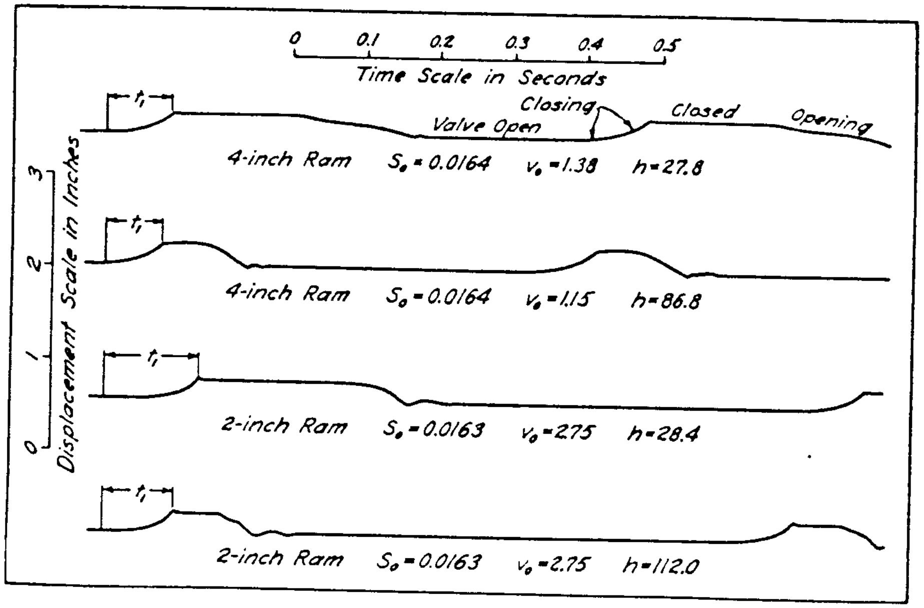


FIG. 15. DISPLACEMENT-TIME DIAGRAMS FOR THE WASTE VALVE

After c was determined by steady-flow test, the velocity v_0 was determined by slowly increasing the velocity, and noting by means of a water manometer the pressure head in the valve box just before closure of the valve. The following relation then determined v_0:

$$v_0 = \sqrt{\frac{2g}{c} \times (\text{head for valve closure})}.$$

TABLE 3

RESULTS OF MEASUREMENT OF t_1 FROM DISPLACEMENT-TIME DIAGRAMS FOR WASTE VALVE OF THE 4-INCH RAM

S_0	t_1 (measured)		
	Counterweight to the Left	Counterweight at the Center	Counterweight to the Right
0.0254	0.106	0.106	0.118
0.0228	0.094	0.104	0.095
0.0186	0.081	0.093	0.092
0.0164	0.081	0.077	0.084

TABLE 4

COMPARISON OF CALCULATED AND MEASURED VALUES OF t_1

S_0 ft.	Ram Size	t_1 Calculated from Equation 4 sec.	t_1 Measured sec.	Difference, per cent $\left(\frac{\text{calculated} - \text{measured}}{\text{measured}} \times 100\right)$
0.0269	2-in.	0.116	0.172	−32.6
0.0254	4-in.	0.130	0.109	+19.3
0.0228	4-in.	0.123	0.099	+24.2
0.0214	2-in.	0.104	0.127	−18.1
0.0186	4-in.	0.111	0.089	+24.8
0.0164	4-in.	0.104	0.080	+30.0
0.0163	2-in.	0.090	0.115	−21.7
0.0140	2-in.	0.084	0.088	− 4.5
0.0093	2-in.	0.068	0.068	0

The stiffness modulus of the waste valve, E_r, was determined by mounting the valve on its seat in a testing machine, applying a known load to the valve, and measuring its deflection with an Ames dial reading in thousandths of an inch. The values of total load on the valve were then plotted as ordinates and the corresponding values of deflection as abscissas, as in Fig. 7, and the slope of the straight line which best fitted the observed points was taken as E_r.

The physical dimensions L, L_1, A, and A_v were measured directly. The following formula, given by Gibson[4] for a pipe free to expand laterally and longitudinally was used to determine a:

$$a = \sqrt{\frac{g}{w\left[\frac{1}{K} + \frac{r}{2uE}\left(5 - \frac{4}{\sigma}\right)\right]}}$$

in which

g = acceleration of gravity, ft. per sec. per sec.
w = specific weight of water, lb. per cu. ft.
K = volume modulus of elasticity of water, lb. per sq. ft.
r_s = radius of pipe, ft.
u = thickness of pipe, ft.
E = modulus of elasticity of pipe, lb. per sq. ft.
σ = Poisson's ratio for pipe material.

All of these values are known quantities except r and u which were obtained by measurement.

18. *Pressure-Time Diagrams.*—A number of records of the relation of pressure to time were obtained by measuring the pressure

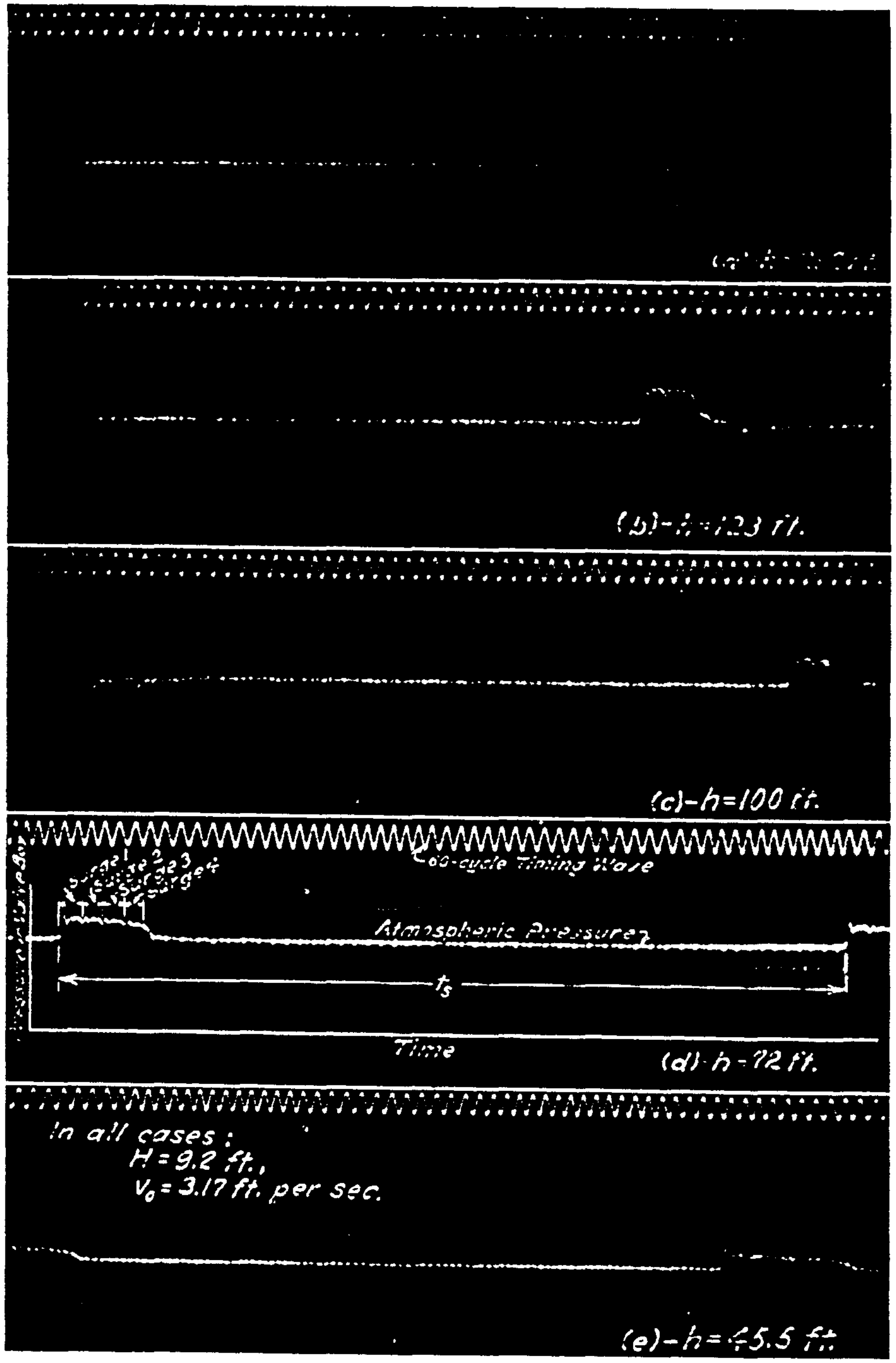


Fig. 16. Pressure-Time Diagrams of the 2-inch Ram

in the drive pipe near the valve box (Fig. 10) with a Peters carbon-pile telemeter, and recording the reading with an oscillograph and traveling film. A few of these records are shown in Fig. 16, and in Fig. 16d the significant portions are labeled. The records are of interest because of their indication of the surges during Period 3, while the check valve was open. The average duration of a surge as indicated on these diagrams was 0.0245 sec.; the calculated value was 0.0251 sec. The quantitative value of the diagrams is questionable, because of the vacuum shown, up to 18 or 20 pounds per square inch, on some of the diagrams (see Fig. 16). It is probable that the shaking of the telemeter caused by the shock of the waste valve closure affected the reading.

19. *Discussion of Results.*—From a study of Fig. 11, it may be seen that an increase of the delivery head decreases the quantity pumped per cycle. Particularly for the 2-in. ram, and to a less noticeable degree for the 4-in. ram, it is evident that the relation between quantity pumped per cycle and delivery head appears to be made up of a series of straight lines rather than a smooth curve. It is also apparent that a decrease of v_0, the velocity required to begin waste valve closure, decreases the quantity pumped per cycle.

From Fig. 12 it may be seen that in general an increase of the delivery head decreases the quantity wasted per cycle. The decrease is greater for the 4-in. ram. For neither ram is the curve smooth; close examination with the aid of Fig. 17 will show that the values of delivery head at which the slope of the Q_s curve is most nearly horizontal correspond rather closely with the values at which the q_s curve changes slope. A decrease of v_0, the velocity required to begin waste valve closure, also decreases Q_s, the quantity wasted per cycle.

From Fig. 13 it may be seen that the duration of the cycle decreases with increasing delivery head, as long as the delivery head is below a certain value at which the duration of the cycle is a minimum; further increase in the delivery head in general increases the duration of the cycle, though not at a constant rate. The changes in duration of cycle are more marked for the 4-in. ram than for the 2-in. ram. From examination of Fig. 17 and the rest of the curves it appears that the values of delivery head at which the slope of the t_s curve becomes nearly horizontal correspond rather closely with the values at which the q_s curve changes slope. A decrease of v_0, the velocity required to begin waste valve closure, also decreases the duration of the cycle, or makes the ram beat faster.

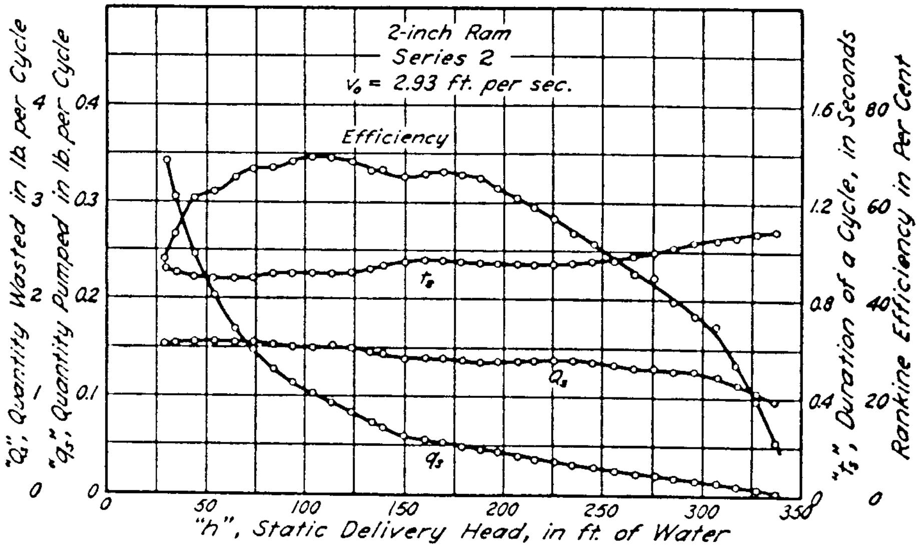


FIG. 17. RELATION OF q_s, THE QUANTITY PUMPED PER CYCLE, Q_s, THE QUANTITY WASTED PER CYCLE, t_s, THE DURATION OF A CYCLE, AND RANKINE EFFICIENCY, TO h, THE STATIC DELIVERY HEAD, SERIES 2, 2-INCH RAM

The variations of the Rankine efficiency, $\left[\frac{q(h-H)}{QH} \times 100\right]$, with delivery head are shown in Fig. 14. It is apparent that there are several maxima for each curve of the 2-in. ram, but that the efficiency curves for the 4-in. ram are smoother. Comparison will show (Fig.

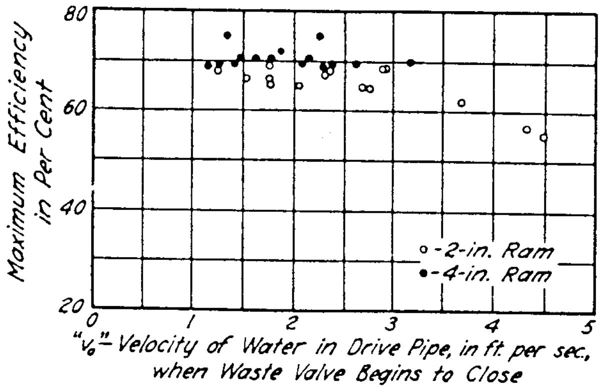


FIG. 18. RELATION OF MAXIMUM EFFICIENCY TO v_0, THE VELOCITY IN THE DRIVE PIPE REQUIRED TO BEGIN WASTE VALVE CLOSURE

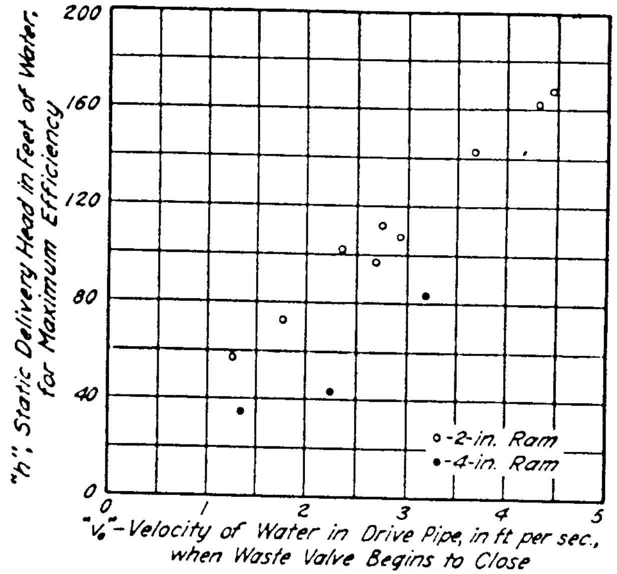


FIG. 19. RELATION OF STATIC DELIVERY HEAD h, AT WHICH MAXIMUM EFFICIENCY IS REACHED, TO v_0, THE VELOCITY IN THE DRIVE PIPE REQUIRED TO BEGIN WASTE VALVE CLOSURE

17) that the maxima of the efficiency curves occur between the values of delivery head at which the q_s curve changes slope.

The foregoing observations, particularly those referring to Fig. 17, are of interest when comparing the experimental results with the calculated performance curves (Section 21).

The variation of the maximum efficiency attained with different values of v_0, the velocity required to begin waste valve closure, is illustrated in Fig. 18. It appears that the maximum efficiency is not much affected by changes in v_0, except for extremely high values of v_0, at which the efficiency is somewhat lower.

The relation of the delivery pressure at which maximum efficiency occurs to the velocity v_0 is shown in Fig. 19. Though the points are somewhat scattered, it may be seen that the pressure at which the maximum efficiency occurs becomes higher as v_0 increases.

IV. COMPARISON OF THEORETICAL AND EXPERIMENTAL RESULTS

20. *Examples of Computations.*—To illustrate the sequence of computations, and to show how the differences in the two rams affected the results, the following examples are shown.

2-in. Ram	Constants	4-in. Ram
3.10	v_0—ft./sec.	3.10
0.0161	S_0—ft.	0.0293
Left	Wt. Position	Left
4.0	J—ft./sec.^{2}	3.0
15.5	j—(dimensionless)	15.5
9.2	H—ft.	9.2
65	h—ft.	65
54.8	L—ft.	55.5
55.8	L_1—ft.	56.5
817	m—ft./sec.	96
3 870 000	E_v—lb./ft.	500 000
0.0233	A—ft.^{2}	0.0884
0.1043	A_v—ft.^{2}	0.371
4 450	a—ft./sec.	4 380
570 000	Y—1/ft.^{2}	58 200
0.00104	Z—sec.	0.0264

2-in. Ram		4-in. Ram
$t_1 = \sqrt{\frac{2 \times 0.0161}{4.0}} = 0.090 \text{ sec.}$	Equation (4)	$t_1 = \sqrt{\frac{2 \times 0.0293}{3.0}} = 0.140 \text{ sec.}$
$\alpha_4 = \frac{\frac{2 \times 32.2 \times 9.2}{15.5} - (3.10)^2}{\frac{2 \times 54.8}{15.5}} = 4.04 \text{ ft./sec.}^2$	Equation (57)	$\alpha_4 = \frac{\frac{2 \times 32.2 \times 9.2}{15.5} - (3.10)^2}{\frac{2 \times 55.5}{15.5}} = 3.99 \text{ ft./sec.}^2$
$v_1 = 3.10 + \frac{4.04 \times 0.090}{2} = 3.28 \text{ ft./sec.}$	Equation (8)	$v_1 = 3.10 + \frac{3.99 \times 0.140}{2} = 3.38 \text{ ft./sec.}$
$Q_1 = 62.4 \times 0.0233 \left[3.10 \times 0.090 + \frac{4.04 \times (0.09)^2}{3}\right] = 0.42 \text{ lb./cycle}$	Equation (10)	$Q_1 = 62.4 \times 0.0884 \left[3.10 \times 0.14 + \frac{3.99 \times (0.14)^2}{3}\right] = 2.54 \text{ lb./cycle}$
$\Delta v = \frac{4 \times 32.2(65 - 9.2) + 817 \times 3.28}{4 \times 4450 + 817} = 0.530 \text{ ft./sec.}$	Equation (35)	$\Delta v = \frac{4 \times 32.2(65 - 9.2) + 96 \times 3.38}{4 \times 4380 + 96} = 0.430 \text{ ft./sec.}$
$v_2 = 3.28 - 0.530 = 2.75 \text{ ft./sec.}$	Equation (15a)	$v_2 = 3.38 - 0.430 = 2.95 \text{ ft./sec.}$
$t_2 = 0.00104 \log_e \frac{3.28}{2.75} = 0.0002 \text{ sec.}$	Equation (20)	$t_2 = 0.0264 \log_e \frac{3.38}{2.95} = 0.0036 \text{ sec.}$
$N = \text{first integer above } \frac{3.28 - 0.53}{2 \times 0.53};\ N = 3$	Equation (37)	$N = \text{first integer above } \frac{3.38 - 0.43}{2 \times 0.43};\ N = 4$
$v_3 = 3.28 - 2 \times 3 \times 0.53 = +0.10 \text{ ft./sec.}$	Equation (44)	$v_3 = 3.38 - 2 \times 4 \times 0.43 = -0.06 \text{ ft./sec.}$

2-in. Ram		4-in. Ram
$v_r = 3.28 - (2 \times 3 - 1)\, 0.53 = 0.63$ ft./sec.	Equation (36)	$v_r = 3.38 - (2 \times 4 - 1)\, 0.43 = 0.37$ ft./sec.
$t_r = 0.00104 \log_e \frac{2 \times 3.28}{2 \times 3.28 - 0.63} = 0.0001$ sec.	Equation (41)	$t_r = 0.0264 \log_e \frac{2 \times 3.38}{2 \times 3.38 - 0.37} = 0.0015$ sec.
$t_3 = \frac{2 \times 55.8}{4450} \times 3 - 0.0002 + 0.0001 = 0.075$ sec.	Equation (42)	$t_3 = \frac{2 \times 56.5}{4380} \times 4 - 0.004 + 0.001 = 0.101$ sec.
$t' = \frac{2 \times 55.8}{4450} - 0.0002 = 0.0249$ sec.	Equation (23)	$t' = \frac{2 \times 56.5}{4380} - 0.0036 = 0.0222$ sec.
$q_e = (62.4 \times 0.0233) \left\{ \begin{bmatrix} 3 \times 3.28 \times 0.0249 \\ + (3 - 1)\, 2 \times 0.00104 \\ \times 0.53 + 0.00104 \times 0.63 \end{bmatrix} - \begin{bmatrix} (3)^2 \times 0.53 \times 0.0249 \\ + (3 - 1)^2 \times 0.53 \times 0.0002 \\ + (3 - 1) \times 3.28 \times 0.0002 \\ + (2 \times 3.28 - 0.63) \times 0.0001 \end{bmatrix} \right\} = 0.185$ lb./cycle	Equation (34)	$q_e = (62.4 \times 0.0884) \left\{ \begin{bmatrix} 4 \times 3.38 \times 0.0222 \\ + (4 - 1) \times 2 \times 0.0264 \\ \times 0.43 + 0.0264 \times 0.37 \end{bmatrix} - \begin{bmatrix} 4^2 \times 0.43 \times 0.0222 \\ + (4 - 1)^2 \times 0.43 \times 0.0036 \\ + (4 - 1) \times 3.38 \times 0.0036 \\ + (2 \times 3.38 - 0.37) \times 0.0015 \end{bmatrix} \right\} = 0.914$ lb./cycle
$v_4 = -\sqrt{(0.10)^2 + \frac{4450 \times 0.00104}{54.8} (0.53)^2} = -0.18$ ft./sec.	Equation (50)	$v_4 = -\sqrt{(-0.06)^2 + \frac{4380 \times 0.0264}{55.5} (0.43)^2} = -0.62$ ft./sec.
$t_4 = -\frac{2 \times 0.00104 \times 0.53}{(-0.18 - 0.10)} = 0.004$ sec.	Equation (52)	$t_4 = -\frac{2 \times 0.0264 \times 0.43}{(-0.06 - 0.62)} + \frac{2 \times 56.5}{4380} = 0.059$ sec.
$v_5 = -(-0.18) = 0.18$ ft./sec.	Equation (53)	$v_5 = -(-0.62) = 0.62$ ft./sec.
$t_5 = \frac{2 \times 54.8 \times 0.18}{32.2 \times 9.2} = 0.067$ sec.	Equation (54)	$t_5 = \frac{2 \times 55.5 \times 0.62}{32.2 \times 9.2} = 0.232$ sec.

2-in. Ram		4-in. Ram
$t_6 = \frac{54.8}{15.5 \sqrt{\frac{2 \times 32.2 \times 9.2}{15.5}}} \log_e \left\{ \frac{\frac{\sqrt{\frac{2 \times 32.2 \times 9.2}{15.5}} + 3.10}{\sqrt{\frac{2 \times 32.2 \times 9.2}{15.5}} - 3.10}}{\frac{\sqrt{\frac{2 \times 32.2 \times 9.2}{15.5}} + 0.18}{\sqrt{\frac{2 \times 32.2 \times 9.2}{15.5}} - 0.18}} \right\} = 0.595$ sec.	Equation (56)	$t_6 = \frac{55.5}{15.5 \sqrt{\frac{2 \times 32.2 \times 9.2}{15.5}}} \log_e \left\{ \frac{\frac{\sqrt{\frac{2 \times 32.2 \times 9.2}{15.5}} + 3.10}{\sqrt{\frac{2 \times 32.2 \times 9.2}{15.5}} - 3.10}}{\frac{\sqrt{\frac{2 \times 32.2 \times 9.2}{15.5}} + 0.62}{\sqrt{\frac{2 \times 32.2 \times 9.2}{15.5}} - 0.62}} \right\} = 0.522$ sec.
$Q_6 = \frac{62.4 \times 0.0233 \times 54.8}{15.5} \log_e \left\{ \frac{\frac{2 \times 32.2 \times 9.2}{15.5} - 0.18^2}{\frac{2 \times 32.2 \times 9.2}{15.5} - 3.10^2} \right\} = 1.48$ lb./cycle	Equation (58)	$Q_6 = \frac{62.4 \times 0.0884 \times 55.5}{15.5} \log_e \left\{ \frac{\frac{2 \times 32.2 \times 9.2}{15.5} - 0.62^2}{\frac{2 \times 32.2 \times 9.2}{15.5} - 3.10^2} \right\} = 5.50$ lb./cycle
$t_c = 0.090 + 0 + 0.075 + 0.004 + 0.067 + 0.595 = 0.831$ sec.	Equation (59)	$t_c = 0.140 + 0.004 + 0.101 + 0.059 + 0.232 + 0.522 = 1.058$ sec.
$Q_c = 0.42 + 1.48 = 1.90$ lb./cycle	Equation (60)	$Q_c = 2.54 + 5.50 = 8.04$ lb./cycle
$Q = \frac{60 \times 1.90}{0.831} = 137$ lb./min.	Equation (61)	$Q = \frac{60 \times 8.04}{1.058} = 456$ lb./min.
$q = \frac{60 \times 0.185}{0.831} = 13.4$ lb./min.	Equation (62)	$q = \frac{60 \times 0.914}{1.058} = 51.8$ lb./min.

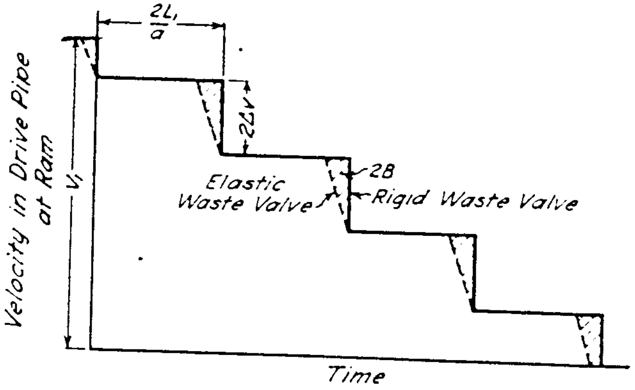


FIG. 20. ASSUMED RELATIONS OF VELOCITY IN THE DRIVE PIPE AT RAM TO TIME DURING PERIOD 3

It is evident from the foregoing examples that evaluation of q_e by the use of Equation (34) is rather tedious. However, in many cases the simplified form given in Equation (43) is sufficiently accurate. Some idea of when the simplified form is permissible may be obtained by noting that if the waste valve is assumed incompressible, the velocity–time curve during Period 3 becomes a series of vertical and horizontal lines, as shown in Fig. 20. In the analysis of Period 3, (Section 11) it has been noted that the area under this curve is proportional to q_e. The difference in area between the two curves for rigid and elastic waste valves is therefore proportional to the error (which is always positive) introduced by considering the waste valve to be rigid. Thus, approximately (see Fig. 20)

$$\text{error in } q_e = 2NwAB \text{ pounds.}$$

But, from Equations (28) and (20),

$$B = v_1 Z \log_e \frac{v_1}{v_2} - Z\Delta v = Z \text{ (function of } v_1 \text{ and } \Delta v).$$

Substitution of this value of B in the equation gives

$$\text{error in } q_e = 2NwAZ \text{ (function of } v_1 \text{ and } \Delta v).$$

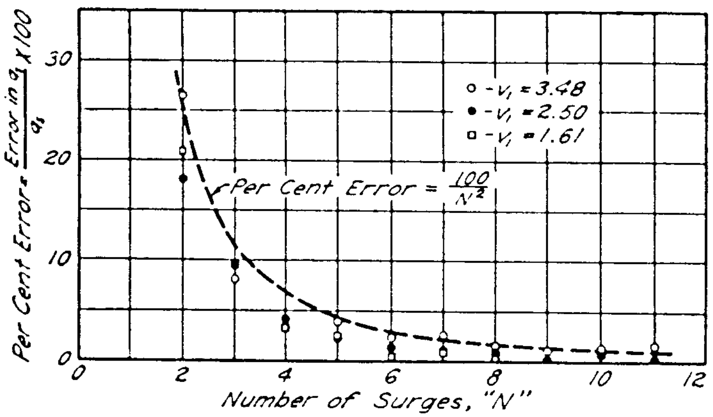


FIG. 21. RELATION OF PERCENTAGE ERROR IN q_e CAUSED BY USING EQUATION (43) TO NUMBER OF SURGES DURING PERIOD 3 FOR THE 4-INCH RAM

$$\text{But percentage error in } q_e = \frac{\text{error in } q_e}{q_e} \times 100$$

$$= \frac{2NwAZ \text{ (function of } v_1 \text{ and } \Delta v)}{\frac{2L_1Aw}{a}(Nv_1 - N^2\Delta v)} \times 100,$$

$$\text{or percentage error in } q_e = \frac{aZ}{L_1} \text{ (function of } N, v_1, \text{ and } \Delta v).$$

For the 4-in. ram the percentage errors have been computed, and the results plotted in Fig. 21 with percentage error as ordinates and N as abscissa. It appears that changes in Δv or v_1 affect the percentage error much less than changes in N, so that the foregoing equation may be written

$$\text{percentage error} = \frac{aZ}{L_1} \text{ (function of } N).$$

The value of $\frac{aZ}{L_1}$ for the 4-in. ram is approximately 2; therefore the function of N may be approximated by the expression $50/N^2$. Then, for any ram,

$$\text{percentage error} = \frac{50aZ}{L_1N^2}.$$

The criterion for applicability of the simplified equation, Equation (43), may be expressed as

$$N \geq \frac{50aZ}{L_1 \text{ (allowable per cent error)}}.$$

For instance, if the allowable error is 5 per cent,

$$N \geq \frac{50 \times 4380 \times .0264}{56.5 \times 5} = 4.52$$

for the 4-in. ram, and the conclusion is drawn that for the 4-in. ram, Equation (43) may be used to calculate q_s, with an error of less than 5 per cent if N is equal to or greater than 5.

21. *Agreement of Results.*—Results, computed by the foregoing method, show that, for all test ranges of operation, the quantity of water pumped per cycle plotted as a function of delivery head becomes a series of sensibly straight lines, intersecting at the values of delivery head at which N, the number of surges in Period 3, changes. Figure 22, on which both computed and experimental values are plotted, shows a good agreement between the values of delivery head at which the computed changes in slope occur, and the pressures at which the actual changes in slope occur.

For the 2-in. ram, the computed value of the quantity pumped per cycle is generally in agreement with experimental results within ten per cent for a range of pressures from the minimum operating pressure of the ram to an upper limit which increases with increasing values of v_0, the velocity required to begin waste valve closure, and which is generally above half the maximum operating pressure for any value of v_0. Above this limit the computed value of quantity pumped per cycle is greater than the experimental value. The conclusion is drawn that the analysis neglects to take into account an energy loss which becomes important at high delivery pressures. One possible explanation of such loss is found in the nature of the setting of this ram. At high pressures, when the waste valve closed, a movement of the entire ram and drive pipe took place, of the order of an eighth of an inch. This was made possible by the flexibility of the sheet metal tank to which the supply end of the drive pipe was attached and by the ram not being attached to the floor

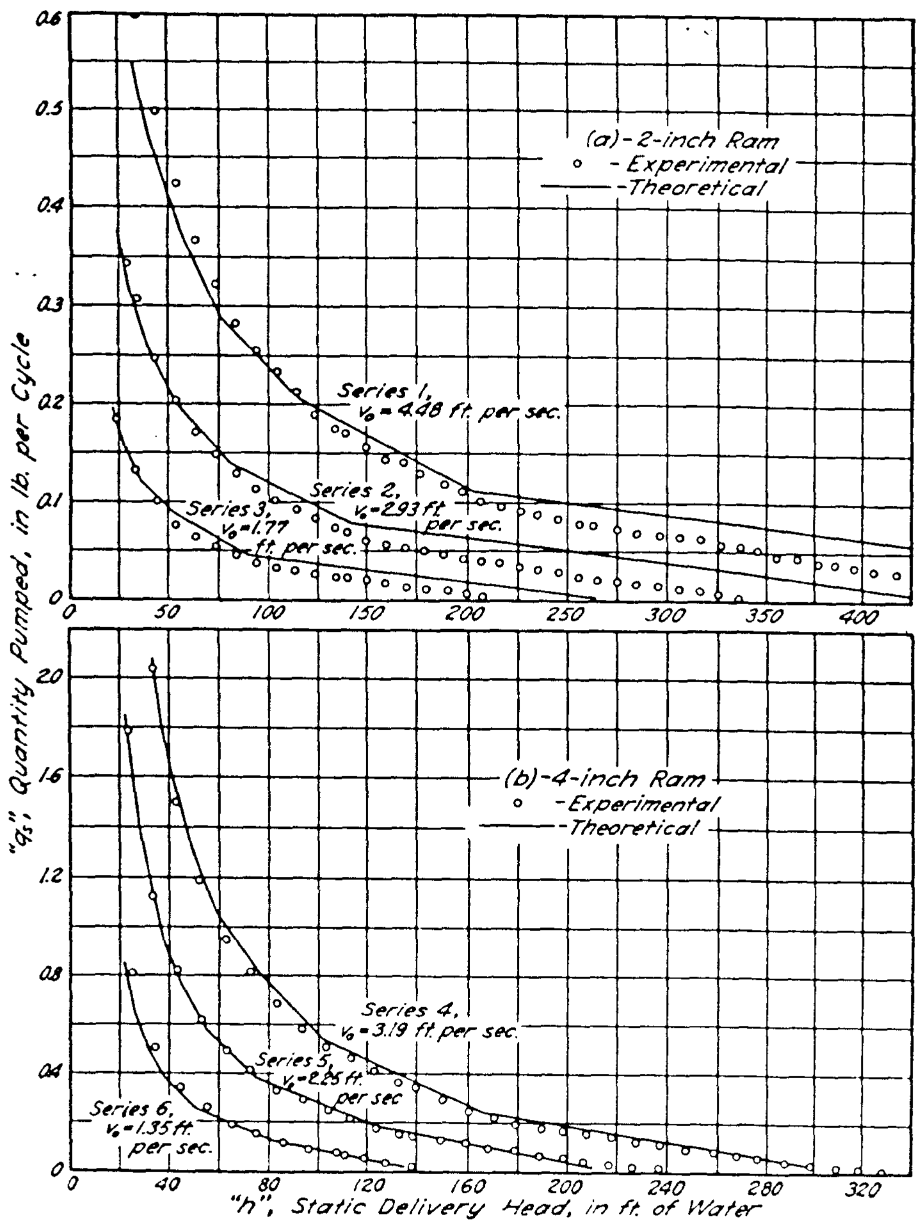


FIG. 22. COMPARISON OF THEORETICAL AND EXPERIMENTAL RELATIONS OF q_s, THE QUANTITY PUMPED PER CYCLE, TO h, THE STATIC DELIVERY HEAD, FOR THE 2-INCH RAM AND THE 4-INCH RAM

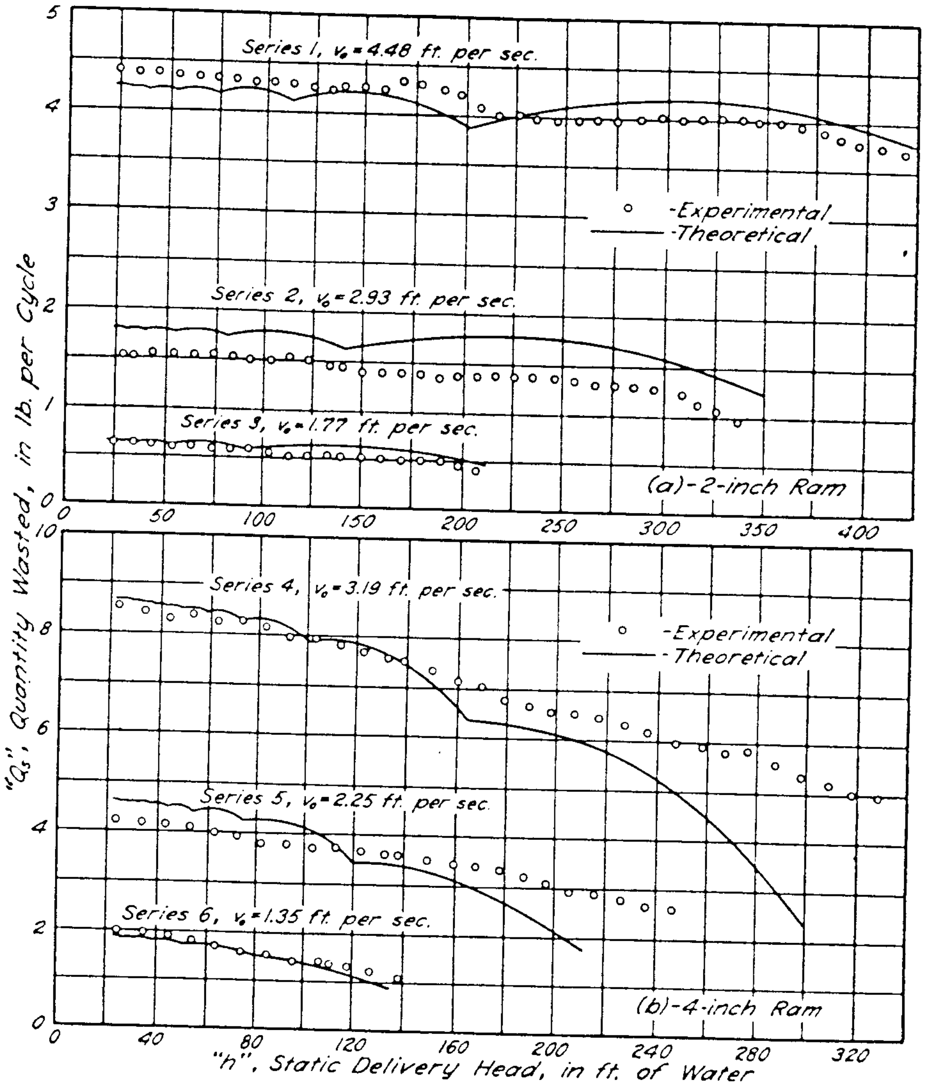


FIG. 23. COMPARISON OF THEORETICAL AND EXPERIMENTAL RELATIONS OF Q_5, THE QUANTITY WASTED PER CYCLE, TO h, THE STATIC DELIVERY HEAD, FOR THE 2-INCH RAM AND THE 4-INCH RAM

of the laboratory. A considerable amount of energy is doubtless dissipated in such motion, but the exact amount is difficult to determine.

For the 4-in. ram the agreement is good throughout the entire range of the tests.

On Fig. 23 are shown the results of computations of the quantity wasted per cycle for the conditions of the six series of tests. It is evident that the general trends of the curves are similar. For the 4-in. ram the curves slope too steeply where the pressure is high; this indicates that the computed values of v_4 or v_5 are too large, since by Equation (58) it is seen that a large value of v_5 results in a small value of Q_6, and therefore a small value of Q_5 (Equation (60)). Probably the assumption that all the energy in the waste valve is returned as the valve expands during Period 4 is responsible for making the computed values of v_4 too large, since the assumption is not true even when the loads are applied slowly (see Fig. 7), and is probably even farther from the truth under faster rates of loading and unloading such as occur in operation of the ram. For the 2-in. ram, a fairly close parallelism, which is most noticeable for the highest value of v_0, is maintained over the entire operating range of the ram.

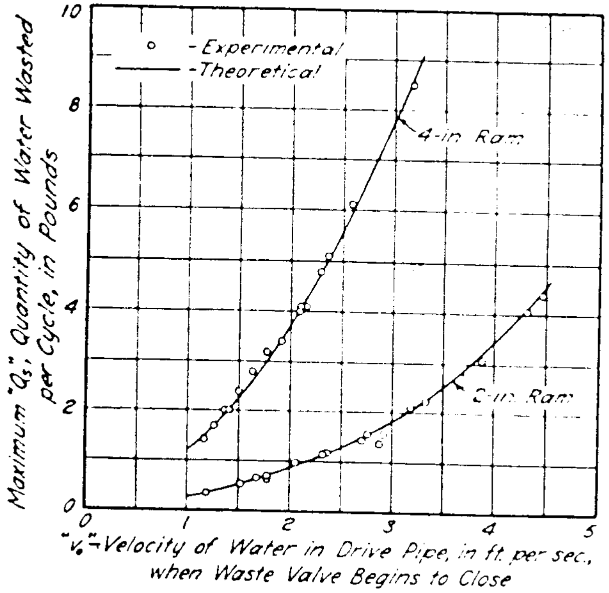


FIG. 24.—COMPARISON OF THEORETICAL AND EXPERIMENTAL RELATIONS OF MAXIMUM Q_5, THE QUANTITY WASTED PER CYCLE, TO v_0, THE VELOCITY IN THE DRIVE PIPE REQUIRED TO BEGIN WASTE VALVE CLOSURE

It is apparent that if the curves were extrapolated to intersect the vertical axis, the intercepts of the computed curves would not all coincide with the values of maximum Q_5 which the experimental curves seem to approach. Figure 24 shows the relation of these intercepts for a number of different values of v_0 for which tests were

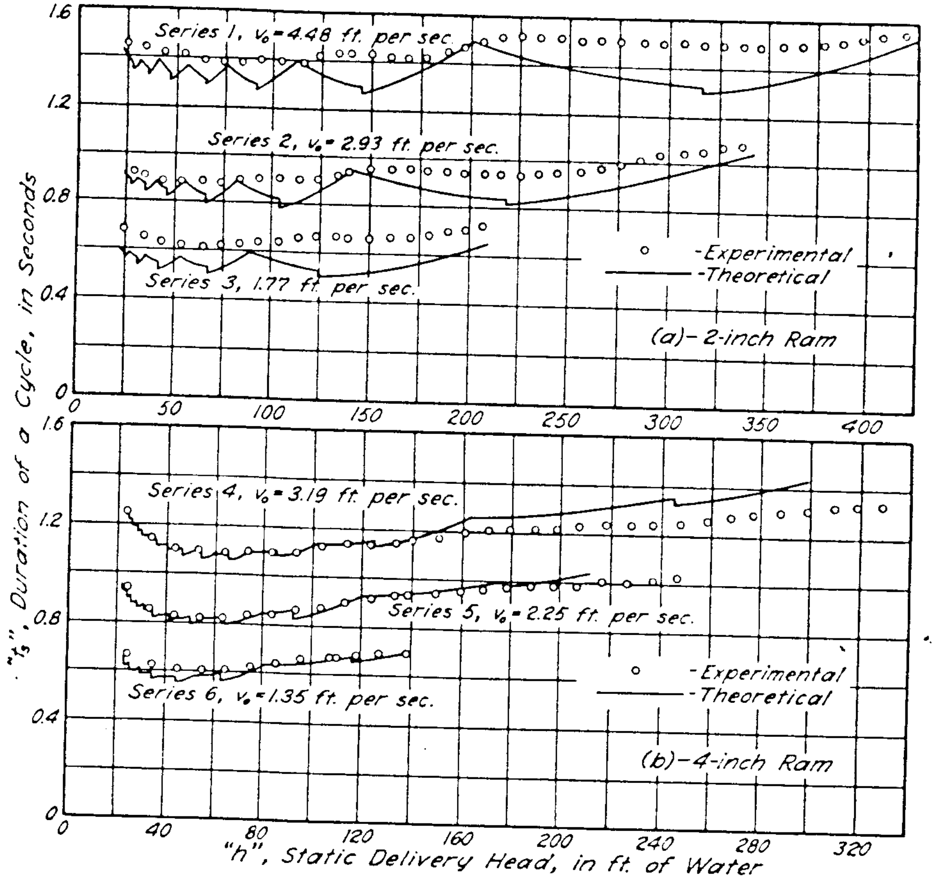


Fig. 25. Comparison of Theoretical and Experimental Relations of t_c, the Duration of a Cycle, to h, the Static Delivery Head, for the 2-inch Ram and the 4-inch Ram

made. From Fig. 24 it may be seen that the method of computing Q_a gives rather good results in general, but that for a few values of v_0 the maximum (intercept) experimental value of Q_a seems a little erratic. It is probable that this is due to inaccurate determination of v_0, which would explain some of the discrepancies noted in connection with Fig. 23.

On Fig. 25 are shown the results of computations for t_c, the duration of one cycle, for the conditions of the six series of tests. Again the general trends of the curves are similar. For the 2-in. ram the experimental points do not follow the dips in the computed curves. The dips in the computed curves are due to the value of v_4 being a minimum at those values of delivery pressure at which v_3 is nearly zero (see Equations (50) and (44)). When the value of v_4 is small, the duration of Period 5 is short (see Equation (54)). Therefore it is probable that v_4 is actually greater than the computed value, for these values of the delivery pressure. The extra energy needed to produce a larger v_4 may come from the recoil from the aforementioned longitudinal movement of the entire apparatus upon waste valve closure.

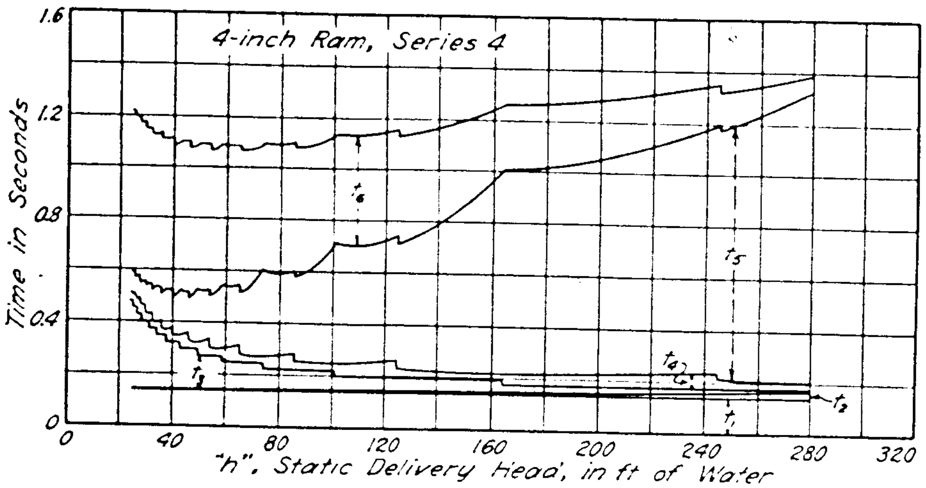


Fig. 26. Variations of t_1, t_2, t_3, t_4, t_5, and t_6, the Durations of Periods 1 to 6, Inclusive, with h, the Static Delivery Head, Series 4, 4-inch Ram

For the 4-in. ram, considerably better agreement is found, except at very high delivery pressures, where, as previously stated, all the energy is not regained from the waste valve as assumed, which makes the computed values of v_4 and t_6 too great.

In Fig. 26 one of the curves for t_c is broken up into its six parts, to show the variation of each with h.

In Fig. 27 computed curves are added to the experimental data of Fig. 17, to show correspondence of the variation in the several curves.

V. Conclusions

22. *Summary and Conclusions.*—It was the purpose of this investigation to make a rational mathematical analysis of the operation of single-acting hydraulic rams and to compare the results of such an analysis with those obtained from experimental data. It was hoped that the analysis would make it possible to calculate satisfactorily the useful characteristics of the ram, especially the

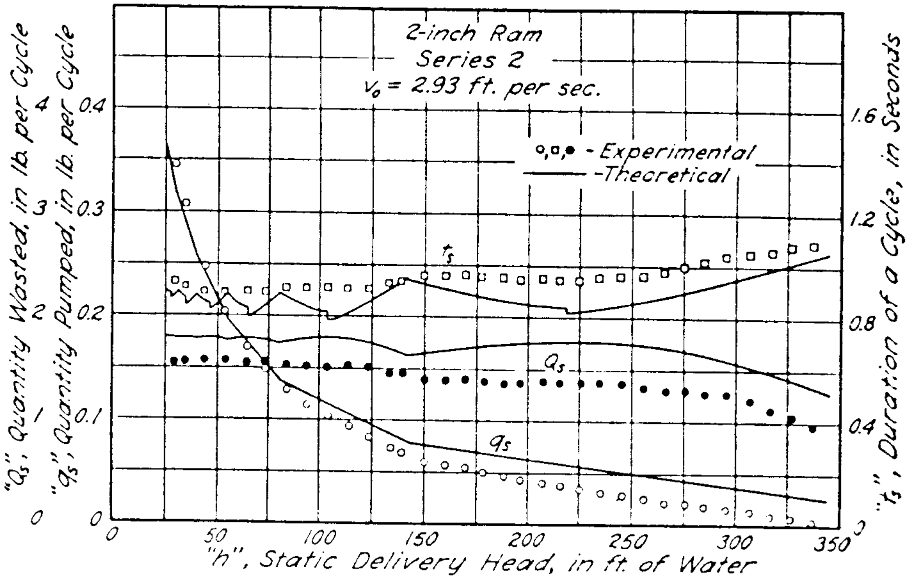


Fig. 27. Comparison of Theoretical and Experimental Relations of q_s, the Quantity Pumped per Cycle, Q_s, the Quantity Wasted per Cycle, and t_c, the Duration of a Cycle, to h, the Static Delivery Head, Series 2, 2-inch Ram

rate of pumping and of wasting for a given supply head, delivery pressure, and waste valve setting. The analysis has been made and is rational except for some minor assumptions which are based on experimental evidence.

Experimental data have been taken on two rams, one having a supply or drive pipe of constant diameter whose nominal value was four inches (designated as a 4-in. ram); the other was a 2-in. ram. The 4-in. ram had a soft waste valve disc, whereas the 2-in. ram had a very hard or rigid waste valve disc. The drive pipe for each ram had a slope of about one ft. in 25 ft. In the mathematical analysis the drive pipe was assumed to be horizontal, since the slope of the drive pipe does not affect the results of the analysis. The dominant factor controlling the functioning of the ram is v_0, the velocity in the drive pipe necessary to cause the waste valve to start closing, and its value is fixed by the waste valve setting. For each ram, series of tests were made with three different values of v_0. In each series of tests the delivery pressure was varied in approximately equal increments over the entire range of pressures under which the ram would continue to operate, all other factors being held constant. The quantity pumped, the quantity wasted, the supply head, the delivery pressure, and the time of the cycle were carefully determined for all series of tests.

A comparison of the results of the mathematical analysis with the results of the experimental data taken from the two rams of different sizes and considerably different characteristics for values of the velocity in the drive pipe covering a wide range of operation appears to justify the following conclusions:

1. (a) The quantity pumped *per cycle* and the quantity wasted *per cycle* can be predicted with an error less than 10 per cent for delivery pressures less than one-half the maximum pressure the ram is capable of developing for a particular setting of the waste valve. The analytical and experimental results are shown graphically in Figs. 22 and 23. For delivery pressures greater than one-half the maximum obtainable, the quantity pumped and quantity wasted may be predicted with somewhat greater error, as indicated in Figs. 22 and 23. However, at these higher delivery pressures the quantity pumped is so small that in general the use of the ram under such conditions is not practical. The correlation between the mathematical and the experimental results is better for the larger ram, particularly for the higher delivery pressures.

(b) The quantity pumped and the quantity wasted *per unit of time* may be computed from the quantities pumped and wasted per cycle if the length of time required for a cycle can be determined. This time for a cycle may be computed with an error less than 10 per cent for the 4-in. ram and less than 20 per cent for the 2-in. ram, as is indicated in Fig. 25.

2. The analysis was based on the fact that pumping was performed by several rapid pressure waves or impulses of the water in the drive pipe during the pumping period in each cycle (designated as Period 3 in the analysis), and not by one surge as the sound of the operating ram would seem to indicate. These pressure waves are indicated on the records of the oscillograph in Fig. 16. This phenomenon is also revealed by the series of straight-line portions of the curves that show the relation between the quantity pumped per cycle and the delivery pressure, as indicated in Fig. 11.

3. In the mathematical analysis involving the pressure-wave phenomenon, the elasticity of the waste valve cannot be neglected for these rams as is evidenced by the marked difference in the performance curves for the two rams as shown in Figs. 11 to 13. (The 2-in. ram was equipped with a much stiffer valve than was the 4-in. ram.)

4. Some of the more important operating characteristics of the single-acting hydraulic rams tested were found to be as follows:

(a) An increase in the delivery pressure or head decreased the quantity pumped per cycle, as indicated in Fig. 11.

(b) An increase in the delivery head decreased the quantity wasted per cycle as indicated in Fig. 12. This decrease was more pronounced in the larger ram.

(c) The time of the cycle decreased as the delivery head increased up to a certain value of the delivery head, (approximately 50 ft. for the 4-in. ram, and 75 ft. for the 2-in. ram); at this head the time of the cycle was a minimum. Further increase in the delivery head increased the time of the cycle as shown in Fig. 13.

(d) The delivery head for which maximum efficiency was attained increased as v_0 increased, v_0 being the velocity in the drive pipe necessary to begin closure of the waste valve.

(e) The maximum efficiency attained varied little with various adjustments of the waste valve, except perhaps for extremely high values of v_0, at which the efficiency was somewhat lower.

APPENDIX

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†Copies of the complete list of publications can be obtained without charge by addressing the Engineering Experiment Station, Urbana, Ill.

Nature. January 21, 1967. p. 318-319.

Reciprocating-jet Pump

A PUMP designed to operate with corrosive liquids at extremely high temperatures will usually be constructed from refractory oxides and hence, if possible, must not rely on solid moving parts such as valves, pistons and rotors. A simple method of achieving this is to make use of some non-reciprocal hydrodynamic effect.

A conventional jet pump is valveless and uses a fluid jet separately pumped into a suitably shaped main pipe, which carries the fluid to be pumped. The main body of fluid is propelled along this pipe by a momentum transfer and mixing process. The injected fluid must, however, be suitable for pumping by conventional means, be compatible with the pumped fluid, and be acceptable as a diluent. At high temperatures these conditions will usually eliminate all fluid combinations apart from a gas or vapour to pump a liquid, and vice versa. Unfortunately there is then a large mismatch in density which causes considerable energy loss during momentum mixing.

Another possibility is to extract from the main pipe an appropriate amount of the fluid to be pumped, in readiness for re-injection. During this extraction or "suck" stroke the fluid mass removed at a sink or jet orifice contains very little momentum before entering the orifice. During the injection or "blow" stroke, however, the jet orifice acts as a source of momentum directed as desired. This is then coupled to the surrounding fluid in the main pipe with an unavoidable but not excessive energy loss. Over a complete "suck–blow" cycle a net forward thrust is created, which may be used to generate a pressure rise or a velocity increase in the main body of fluid, depending on the shape chosen for the main pipe.

The non-reciprocal action of an alternating jet depends on the fact that the flow pattern for a steady suck stroke is fundamentally different from that for a steady blow stroke. This may be seen in the streak photograph (Fig. 1), which was taken in a two dimensional water tank using polystyrene beads as markers. The left-hand jet is sucking and the right-hand blowing. During the suck stroke irrotational potential flow occurs with a convergent "scalar" flow pattern in which $\mathrm{curl}\,\mathbf{v} = 0$, where $\mathbf{v}$ is the fluid velocity. At the start of the blow stroke a divergent flow pattern occurs momentarily, but because of viscous forces in the boundary layers the liquid cannot sustain the large velocity gradients around the edges of the jet orifice, and a rotational motion is started. The transient potential flow pattern therefore breaks down as inertial forces take over control. Ring vortices are generated around the periphery of the jet orifice and are shed downstream, so that in the steady state there is a gradual erosion of the jet associated with a widening of its influence as ejected fluid enters a turbulent mixing region. A steady thrust is the ultimate result. The flow pattern of the blow stroke might be regarded as a "vector" pattern in which $\mathrm{curl}\,\mathbf{v} \neq 0$.

The problem of pumping a steady stream of fluid has now been transformed to that of cycling fluid through jet orifice. In a severe environment the latter problem is much more simple to solve, for example, by use of a vertically reciprocating gas–liquid interface stabilized by gravity. This forms a piston in a chamber coupled to the jet orifice by means of a suitable diffuser or nozzle. In the case of a pump for liquids, the gas pressure above the liquid surface may be cyclically varied by means of a reciprocating mechanical piston or by a valve-controlled supply of compressed gas, so as to blow and suck alternately through the jet orifice. Two alternative methods of varying the gas pressure are based on the two-stroke internal-combustion engine and the steam engine. In the latter method, it might be advantageous to use the vapour of the liquid being pumped. A more speculative technique, which is particularly suited to the pumping of some very hot liquids, would be to inject water on to or below the surface of the hot liquid within the piston chamber. This water would be converted immediately to steam, forcing liquid out through the jet. The piston chamber could then be opened to a condenser for the suck stroke. The pumping power would be derived at the expense of a small drop in liquid temperature.

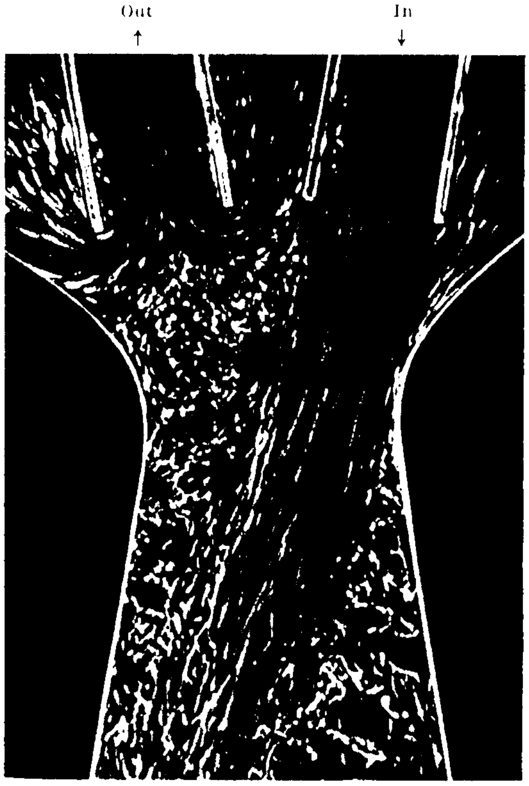

Fig. 1. Reciprocating-jet pump.

Experiments have been carried out on the pumping of water in various configurations of pump, using up to four jets. Particular attention is now being given to "twin-jet" forms of the pump in which each stage comprises two jets in anti-phase directed at a venturi orifice. The streak photograph (Fig. 1) shows that there is little interference between the two jets. An experimental pump, with angled jets attached to the main pipe like the branches of a tree, has been constructed from drilled refractory bricks and will be tested with molten salts at 1,500° K.

A simple theory, based on separate consideration of steady state blowing and steady state sucking, has been found to give reasonable agreement with the results. The pump head (h_P) in m or ft. of the liquid pumped is given for equal periods of blow and suck by:

$$h_P = [n\alpha\overline{v_J^2}/2g][1 - 2(v_P/\bar{v}_J) - \alpha/2]$$

where n is the total number of jets; α is the ratio of the area of each jet to that of the pipe; $\bar{v}_J$ is the mean jet velocity; $\overline{v_J^2}$ is the mean square velocity; v_P is the pipe velocity just before mixing, and g is the acceleration due to gravity. It is assumed that when more than one jet is involved, the jet velocities are suitably phased to give zero net addition of fluid at any given moment.

At high frequencies, or short wavelengths, as defined by the ratio of pipe velocity to frequency, the steady state treatment is likely to be less accurate. In order to allow for transients during switch-over, a more sophisticated approach is required which involves considering the ring vortex streets.

The main drawback to the reciprocating-jet pump is that the pumping efficiency is unlikely to exceed 38 per cent at best, and a more realistic figure would probably be half this. Nevertheless there are many applications where efficiency is less important than some other consideration such as the absence of solid moving parts. The reciprocating-jet pump should be considered for application to the pumping of the following materials: molten metals, slags, glasses and salts, where high temperatures and corrosion are a problem; blood, where contamination must be avoided; solid suspensions, where blockage and erosion can occur, and liquid gases such as helium at low temperatures.

My thanks are due to Mr. J. R. Stutely for many useful ideas in development of the pump and also to Mr. R. C. Kell for valuable comments on the manuscript.

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GLANDLESS RECIPROCATING-JET PUMP

1. Introduction

A pump designed to operate with corrosive liquids at extremely high temperatures will usually be constructed from refractory oxides, and hence, if possible, must not rely on solid moving parts such as valves, pistons and rotors. A simple method of achieving this is to make use of some non-reciprocal hydrodynamic effect.

A conventional jet pump is valveless and uses a fluid jet separately pumped into a suitably shaped main pipe, which carries the fluid to be pumped. The main body of fluid is propelled along this pipe by a momentum transfer and mixing process. The injected fluid must, however, be suitable for pumping by conventional means, be compatible with the pumped fluid, and be acceptable as a diluent. At high temperatures these conditions will usually eliminate all fluid combinations apart from a gas or vapour to pump a liquid, and vice versa. Unfortunately there is then a large mismatch in density which causes considerable energy loss during momentum mixing.

2. Principles of Operation

The technique used in the reciprocating-jet pump is to extract from the main pipe an appropriate amount of the fluid to be pumped, in readiness for re-injection. During this extraction or "suck" stroke the fluid mass removed at a sink or jet nozzle contains very little momentum prior to entering the nozzle. During the injection or "blow" stroke, however, the jet nozzle acts as a source of momentum directed as desired. This is then coupled to the surrounding fluid in the main pipe, with an unavoidable but not excessive energy loss. Over a complete "suck blow" cycle a net forward thrust is created, which may be used to generate a pressure rise or a velocity increase in the main body of fluid, depending on the shape chosen for the main pipe. A similar principle has been studied for use in ship propulsion.

The non-reciprocal action of an alternating jet depends upon the fact that the flow pattern for a steady suck stroke is fundamentally different from that for a steady blow stroke. This is shown in Fig. 1 and has been observed in a two-dimensional water tank using polystyrene beads as markers. The left-hand jet is sucking and the right-hand one blowing. During the suck stroke irrotational potential flow occurs with a convergent "scalar" flow pattern in which curl $\mathbf{v} = 0$, where $\mathbf{v}$ is the fluid velocity. At the start of the blow stroke a divergent flow pattern occurs momentarily, but because of viscous forces in the boundary layers the liquid cannot sustain the large velocity gradients around the edges of the jet nozzle, and a rotational motion is initiated. The transient potential flow pattern therefore breaks down as inertial forces take over control. Ring vortices are generated around the periphery of the jet nozzle and are shed downstream, so that in the steady state there is a gradual erosion of the jet associated with a widening of its influence as ejected fluid enters a turbulent mixing region. A steady thrust is the ultimate result. The flow pattern of the blow stroke might be regarded as a "vector" pattern in which curl $\mathbf{v} \neq 0$.

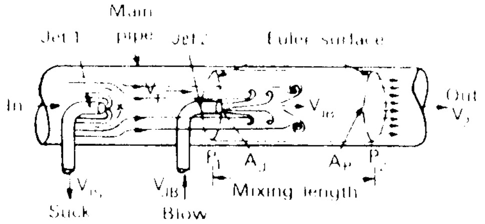

Fig. 1. Simple pump showing "suck" and "blow" patterns.

The problem of pumping a steady stream of fluid has now been transformed to that of cycling fluid through a nozzle. In a severe environment the latter problem is much more simple to solve, for example by use of a vertically-reciprocating gas liquid interface stabilized by gravity. This forms a piston in a chamber coupled to the jet nozzle by means of a suitable diffuser. In the case of a pump for liquids, the gas pressure above the liquid surface may be cyclically varied by means of a reciprocating mechanical piston or bellows or by a valve-controlled supply of compressed gas, so as to blow and suck alternately through the jet orifice.

3. Some Possible Designs

The simplest form of pump comprises a pipe with injector tubes or jet nozzles arranged at various points along the main pipe of the pump, and directed in the flow direction (Fig. 1). The pipe and jet nozzles are filled with liquid. Suitably phased alternating pressure pulses are applied to the jet nozzles by one of the methods just described. During the blow stroke, the pressure in the nozzle is greater than in the pipe so that liquid flows out to form a jet with "vector" flow along the pipe. During the suck stroke, the pressure is reversed and an equal amount of liquid flows back into the nozzle to produce a convergent or "scalar" flow pattern.

In the first practical design of reciprocating-jet pump (Fig. 2) the jet tubes are fitted to the pipe like

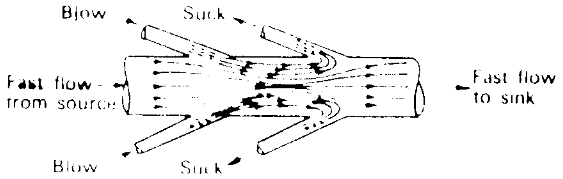

Fig. 2. "Tee" pump for high velocities.

the branches of a tree, with no extension within the pipe. This design has the advantage of simple construction, low resistance to flow in the pipe, and least surface area open to corrosion by the liquid being pumped. The design will be most useful when the requirement is for a high flow rate across a small pressure gradient and when efficiency is of little importance.

The second practical design is the "bulb-venturi" pump, shown in Fig. 3, which includes the air/liquid piston chambers. The pipe is shaped so as to provide a venturi and diffuser mixing chamber. The main purpose of the former is to accelerate the main liquid stream to a velocity nearly matching that of the jet so that less energy is lost during momentum mixing; the diffuser converts the velocity head back to pressure during the mixing. Each stage incorporates two jets cycling in anti-phase, so as to produce a more even thrust and continuity of mass. A larger number of suitably phased jets per stage would produce smoother flow, but space is limited close to the venturi. The jet nozzles have a rounded profile and a 10° diffuser angle so as to reduce losses and circulation during the suck stroke. Streak photographs of two-dimensional flow patterns in a shallow tank have shown that, with the correct geometry, there is negligible interference between the blowing and sucking jets. This design will be most useful in a system requiring low flow rates across large pressure gradients.

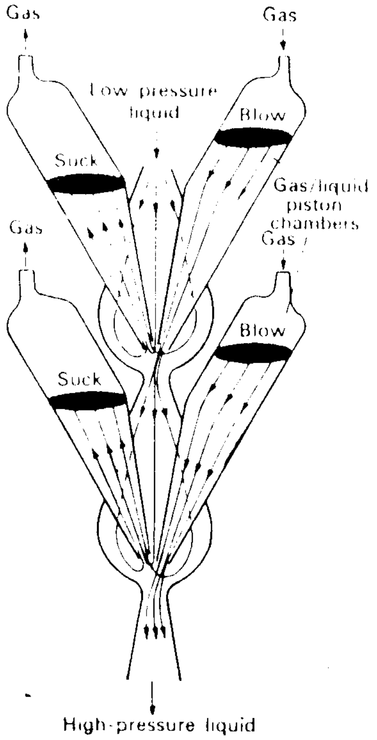

Fig. 3. Two stage twin-jet bulb-venturi pump.

A design, which in section might be similar to Fig. 2 rotated clockwise through 90°, consists basically of a series of similar inverted cones which are concentric with a vertical axis. Each conical cavity thus formed is sealed at the top, with small gas pipe connections, and has a shaped orifice at the bottom directed into the cavity beneath. Initially the top half of each cavity is filled with air and the lower half with liquid which is continuous throughout all cavities. During the first half of a pump cycle, a high air pressure is applied to alternate cavities and a low pressure to the intermediate cavities. Liquid is thus forced into the low-pressure cavities. Owing to the shaping of the orifice and to the creation and ejection of momentum downwards from the high-pressure cavity, liquid is forced preferentially into the cavity below. The second half of the pump cycle involves the reversal of the whole operation in a given cavity, although for the pump as a whole there is a net transfer of liquid downwards against the pressure gradient. Alternative phasing of the "travelling wave" type could be advantageous. This design may be suitable for viscous liquids.

4. Theory

A simple theory, based on separate consideration of quasi-static blowing and sucking, has been found to give reasonable agreement with the results, provided boundary layer corrections are applied. The pump head h_P may be expressed:

$$h_P = nk\bar{v}^2_{JB}\alpha(1 - \alpha/2 - 2v_P/\bar{v}_{JB})/g \tag{1}$$

where n is the number of jets,

k is the ratio of blow period to the period of one cycle,

$\bar{v}^2_{JB}$ is the mean square jet velocity during T_B,

α is the effective area ratio of jet nozzle to main pipe (Fig. 1),

v_P is the liquid velocity in the main pipe,

$\bar{v}_{JB}$ is the mean jet velocity,

g is acceleration due to gravity.

At high frequencies, or short wavelengths as defined by the ratio of pipe velocity to frequency, the steady state treatment is likely to be less accurate. In order to allow for transients during switch-over, a more sophisticated approach might be required which involves consideration of ring vortex streets.

Equation (1) relates pump performance to the jet velocity either as measured, or as programmed by hydraulic coupling. However, it does not consider the way in which the jet velocity was achieved, for example by pneumatic coupling to a mechanical piston with a sinusoidal volume displacement. This has required a more extensive treatment since the drive pressure is related to the results it achieves, so the differential equations are non-linear and must be solved with the help of an analogue computer. If, however, the pressure variation as a function of time is known, either from a recorder or as a result of programming a valve controlled compressed air supply, it is possible to estimate pump performance.

A simple mathematical model may be used to describe the behaviour of two piston chambers attached to nozzle/diffusers, which feed liquid alternately from one to the other via the pump body. Assuming gravitational and inertial effects within the piston chambers to be small, but taking into account typical nozzle/diffuser losses, the mean square of the jet blow velocity available from a twin-jet combination at the rth stage may be expressed:

$$\overline{c^2_{JBr}} = 2C_J \overline{\Delta p_r}/\varrho \qquad (2)$$

where C_J is the nozzle/diffuser loss coefficient (= 0·67 for example),

$\overline{\Delta p_r}$ is the mean of the modulus of the difference in pressure between the two rth piston chambers during an average cycle,

ϱ is the liquid density.

Substituting (2) in (1) for each stage, with $k = \frac{1}{2}$, we have the pump stagnation head ($v_P = 0$) for n twin-jet stages given by:

$$h_{P0} = 2C_J x(1 - x/2)\left[\sum_1^n \overline{\Delta p_r}\right]/\varrho g \qquad (3)$$

Experience has shown that the suck contribution to the pressure difference $\overline{\Delta p_r}$ sometimes has little effect, especially when cavitation occurs. Under these circumstances the use of the mean blow pressure alone in equation (3) gives satisfactory agreement with experiment.*

5. Experimental results

A small hot single-stage twin-jet model, similar in principle to Fig. 3, but using piston chambers curved through 90°, has been constructed from silica glass with a jet diameter of 0·7 cm venturi bore of 1 cm. This has pumped molten potassium sulphate at 1500°K at flow rates up to 0·18 l/sec across a head of 13·5 cm. This corresponds to a stagnation head of 51 cm (6·8 cm Hg). The pump was pneumatically coupled via a pair of water-cooled pipes to two mechanical pistons of 60 cm³ displacement, which were sinusoidally cycled in anti-phase at frequencies in the range 2 to 5 Hz.

A somewhat larger pump for molten aluminium has been constructed from graphite with a design similar to Fig. 2, by A. E. Pengelly and S. Pilkington of British Aluminium Company Limited Research Laboratories. There were four jets of 2 cm diameter fitted to a main pipe of 7·5 cm bore. The drive was by means of compressed air actuated mechanical pistons, which were coupled pneumatically to the piston chambers containing molten aluminium at 1000 K. The pump achieved a stagnation head of ca. 25 cm (5 cm Hg) and should provide a flow rate of 10 l/sec across a 5 cm head.

Water experiments have been carried out using a two stage model closely similar to Fig. 3, with jet and venturi dimensions identical to the silica pump. This pump was driven by four servo-controlled spool-valves switching air pressures of up to 1·3 atm (100 cm Hg). At this drive pressure, a stagnation head of 50 cm Hg (8 cm for the first stage) was achieved at a frequency of 0·86 Hz, without the need for a sub-atmospheric suck stroke. This is because a small static pressure head at the first stage is sufficient to provide a reasonable jet suck velocity when assisted by the dynamic pressure rise in the nozzle/diffuser. A much greater head is produced by the second stage (42 cm), since the suck stroke is helped by the pressure developed by the first stage. The suck stroke of downstream stages is more effective due both to the increased pressure, and to the consequent alleviation of any limitation in suck velocity as a result of cavitation.

Considerable experimental work has been carried out using a pump of similar jet and venturi geometry, but with long straight sided 1·6 cm bore piston chambers driven by the pneumatically coupled mechanical drive described earlier. A stagnation head of 32·5 Hg has been achieved at 2·5 Hz for two stages each with a mean blow pressure of 0·45 atm (34·5 cm Hg).

6. Efficiency

There are basically three efficiencies which require consideration with the reciprocating-jet pump. Firstly there is the hydrodynamic or pumping efficiency, the efficiency of conversion from liquid energy in the air/liquid piston chamber to work done on the pumped liquid. Secondly there is the pneumatic efficiency, the efficiency of conversion of pneumatic energy to liquid energy in the piston chamber, and finally there is the efficiency of gas compression in creating the pneumatic energy.

The main drawback to conventional jet-pumps is that the pumping or hydrodynamic efficiency is not more than 38 per cent mainly as a result of mixing losses. There are, however, two important differences when the reciprocating-jet pump is compared with this. Firstly there is a suck stroke, which consumes power while providing no output, but secondly there is much less restriction on the allowed values of the ratio of jet to venturi area. This ratio could approach unity with a reciprocating-jet, which could result in a considerable increase in efficiency to near that for a diffuser (say 70 per cent).

If a mechanical piston, bellows or alternative closed system is used, the pneumatic efficiency could be near 100 per cent but unfortunately the efficiency with which the pneumatic energy is created is likely to be low. Alternatively compressed gas drive may be used which has been efficiently produced. However, it is likely that some pneumatic energy will be discarded to the exhaust on the suck stroke.

In a hot system the liquid enthalpy could perhaps provide some power by expanding injected cold gas or generating steam, but these possibilities remain to be investigated. It is difficult to generalize, but the overall efficiency is unlikely to be greater than 40 per cent.

7. Applications

There are many applications where efficiency is less important than some other consideration such as the absence of solid moving parts. The reciprocating-jet pump should be considered for application to the pumping of the following materials: molten metals, slags, glasses and salts, where high temperatures and corrosion are a problem; blood, where contamination and damage must be avoided; solid suspensions, where blockage and erosion can occur; and liquid gases such as helium at low temperatures.

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A. J. WALKDEN

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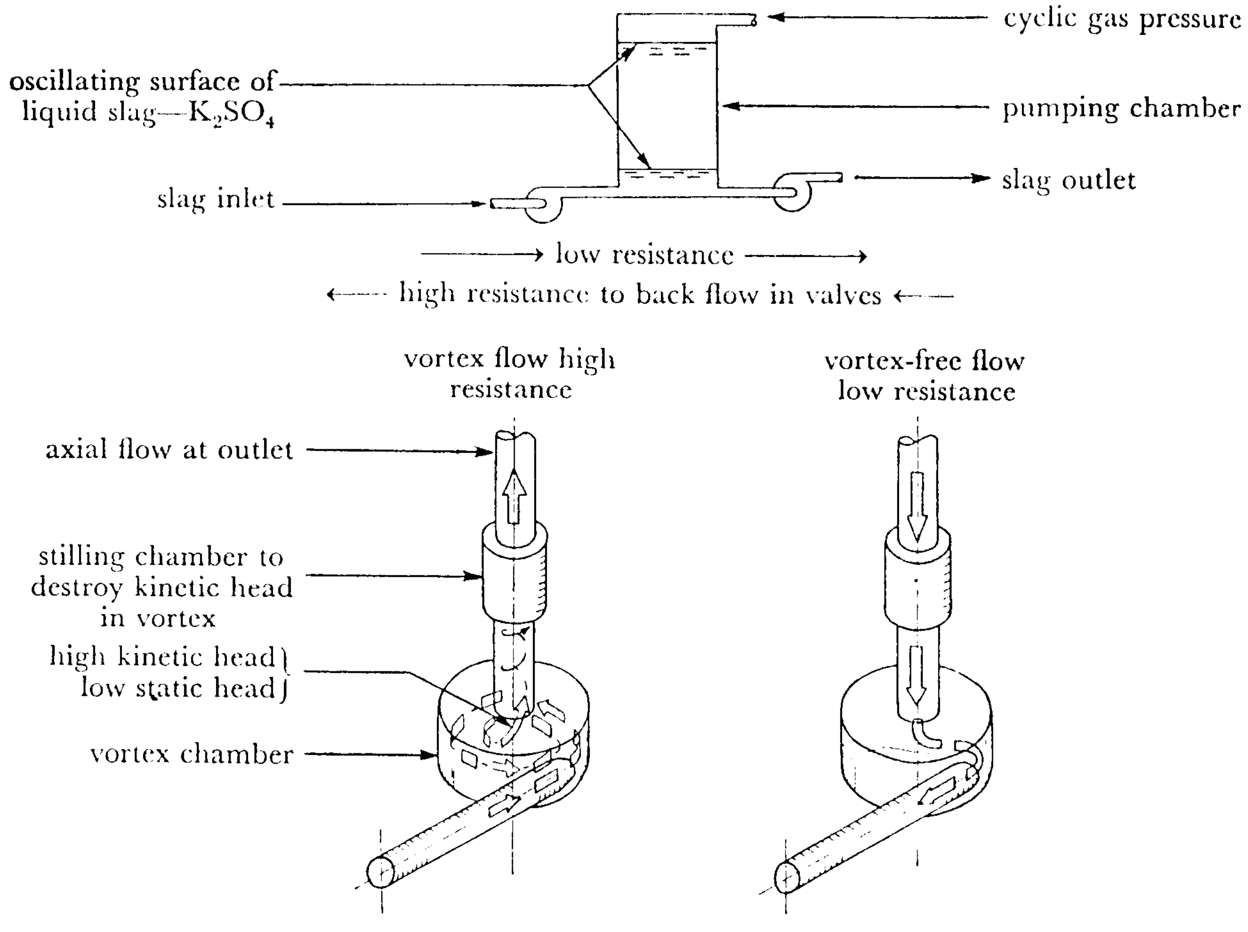
were sufficiently encouraging to warrant slag atomization experiments on a more realistic scale and these will be carried out in the plant shown in figure 11, plate 26. The effect of varying slag temperature over the range 1100 to 1600 °C will be observed as well as the flow and geometrical factors already investigated on the water model. The slag spray will fall under gravity against an upward current of air and the effect of the air stream on the shape of the spray and on the carry-over of fine particles will be observed. It will be possible to make measurements of temperatures and mass flows and obtain an experimental check on the method of design for heat transfer.


cyclic gas pressure
oscillating surface of liquid slag—K_2SO_4
pumping chamber
slag inlet
slag outlet
low resistance
high resistance to back flow in valves
vortex flow high resistance
vortex-free flow low resistance
axial flow at outlet
stilling chamber to destroy kinetic head in vortex
high kinetic head
low static head
vortex chamber

operation of fluid diodes

FIGURE 12. Schematic arrangement of the 'slag' pump using 'fluid diodes' as valves.

2·3·3. *The slag pump*

As has previously been discussed, many of the difficulties arising in the development of a suitable air heater for m.h.d. are due to the pressure difference between the air and the gas from which the heat is to be extracted. In the liquid slag air heater, this pressure difference will support a column of slag of the order of 100 ft. high. Thus, if one wishes to make use of gravitation to transfer the liquid from the bottom of the gas chamber to the top of the air chamber, there must be a vertical separation of approximately 100 ft. between the two chambers. The implications of this in terms of cost of supporting structure and ductwork provide a considerable incentive to develop a pump to provide the necessary transportation of liquid slag.

A survey of existing experience in this field shows that liquid metals have been pumped by centrifugal pumps at temperatures up to 800 °C. In this case, uncooled metal impellers have proved practicable. In the liquid slag regenerator the slag temperature will be at least 1300 °C and may be as high as 1600 °C. Under these conditions, it is not possible to cool the impeller sufficiently well to remove the heat that would be transferred to it by forced convection from the liquid. A coated or ceramic impeller is also impracticable because the only materials able to withstand the chemical attack of potassium sulphate do not have sufficient mechanical strength to resist the centrifugal forces. These considerations apply also to any form of mechanical positive displacement pump.

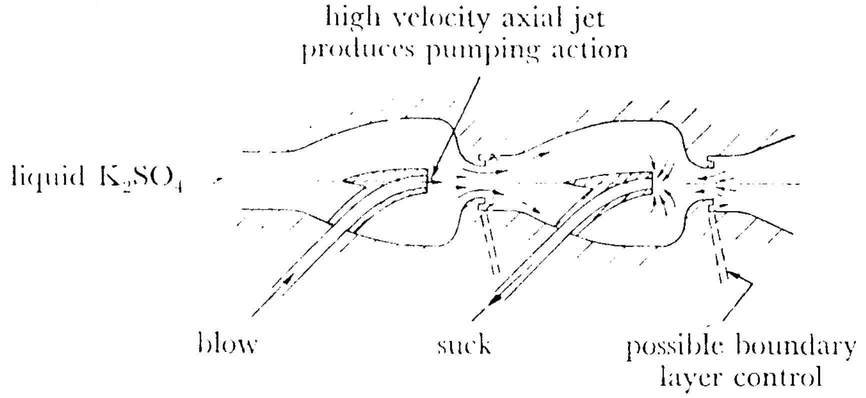


FIGURE 13. Schematic arrangement of the vector flow 'slag' pump

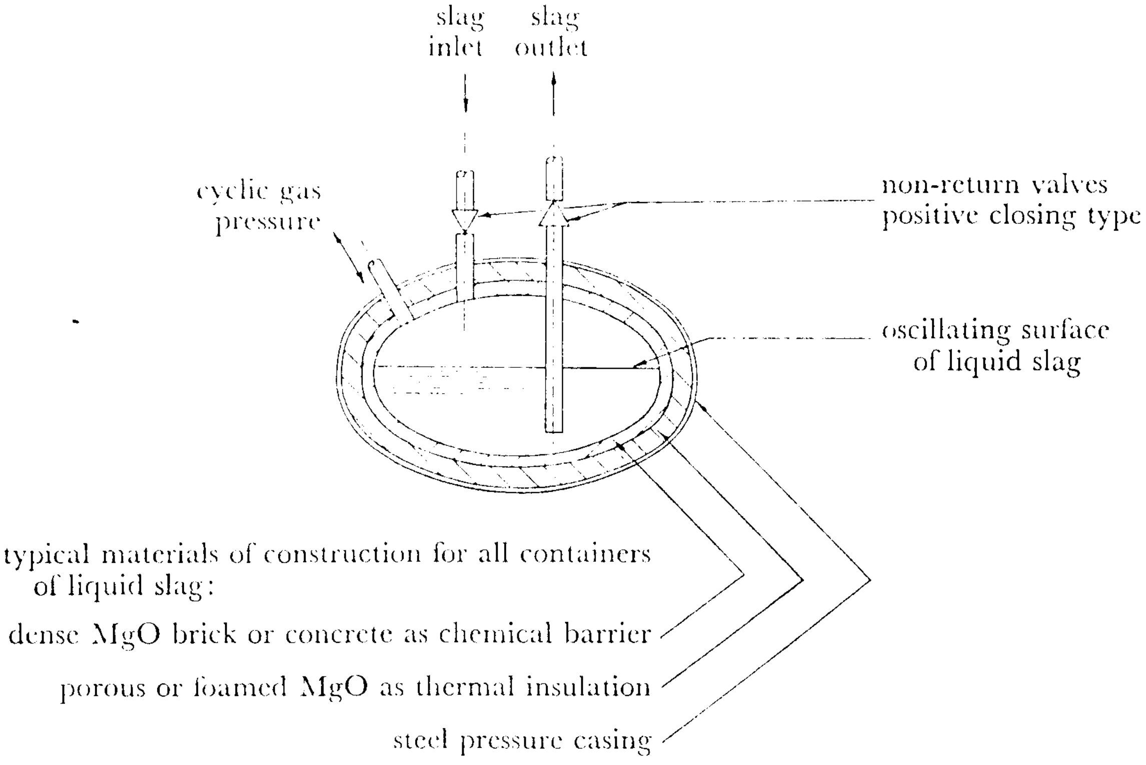


FIGURE 14. Schematic arrangement of the 'acid egg' pump.

Thus, the choice of practicable pump is restricted to a form in which a container is allowed to fill with liquid slag which is then expelled by a gas under pressure. Containment of the liquid is practicable and the gas forms a buffer between the hot liquid and the pumping prime mover.

In this form, there are two principles on which a pump could operate: (*a*) with valves of some sort to constrain the liquid flow in the desired direction—the 'liquid diode' and 'acid egg' pumps; (*b*) by making use of the change of liquid momentum to produce pressure difference—the 'vector pump . The principle of each of these devices all of which are currently under investigation, is illustrated in figures 12, 13 and 14.

C_F, C_B coefficients of resistance for forward and backward flow where $\Delta P = \frac{1}{2}C\rho v^2$

P_o = outlet pressure = 8 atm
P_i = inlet pressure = 2 atm
P_S = suction pressure = 1 atm
P_P = pumping pressure = variable

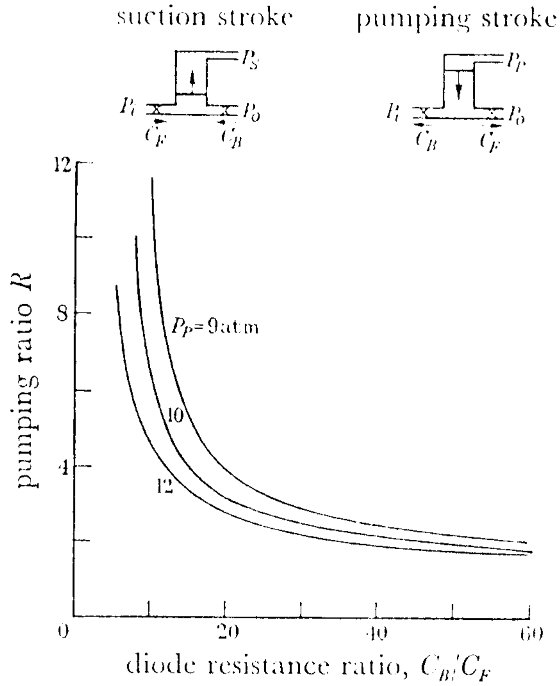


FIGURE 15. Variation of pumping ratio R (volumetric efficiency) with diode resistance ratio C_B/C_F for different pumping pressures.

In the 'liquid diode', the flow of liquid in one direction produces a vortex which is subsequently removed in a stilling chamber, whereas flow of liquid in the opposite direction does not produce a vortex. The flow of liquid in the vortex-producing direction is accompanied by an energy loss due to the vortex and therefore the resistance to flow is greater in this direction than in the direction where no vortex is generated. The initial experimental work is being carried out using Perspex models and water as the liquid. With the first model, a backward to forward resistance ratio of 23:1 has been measured and when a better understanding of the way in which the geometry of the vortex chamber is reached, it is hoped that higher resistance ratios will be achievable. Figure 15 shows the pumping ratios, i.e. the number of pumping cycles needed to transfer one swept volume plotted against liquid diode resistance ratio for various operating conditions.

The 'acid egg' pump requires the use of moving parts in contact with the slag. This type of pump is used in the chemical industry to handle corrosive fluids but its use is so far restricted to low temperatures. Design studies are proceeding on a high temperature version but selection of a suitable material must first be made.

The 'vector pump' makes use of the fact that liquid drawn from a reservoir into a jet has a multidirectional flow, whereas when it is ejected from the jet is has a unidirectional flow. It has been demonstrated that reciprocating operation of a single jet produces better pumping action than was originally expected from consideration of suction and ejection separately. Four elements arranged in series and suitably phased give five times the head produced by a single element.

The development of a suitable pump is considered vital to the economic application of the slag air heater principle. Although the experimental results so far achieved give some encouragement in so far as small pressure rises have been achieved, they should be treated with extreme caution until multiple arrays giving the complete pressure rise have been tested and it is also shown that the critical geometries can survive in high temperature conditions.

2·3·4. *Pellet distribution into upper chamber*

Particles of nominally uniform size must be distributed uniformly over the horizontal cross-section of the chamber at a constant feed rate. Feeders with moving parts are unacceptable due to the high temperature and the corrosive nature of the material. A number of gravity feed systems have been considered and it has been shown that a simple flat perforated plate is suitable. The minimum size of the perforations, to avoid blockage by the ⅛ in. nominal diameter particles, has been shown to be a ¾ in. diameter hole. Measuring the mass flow rate of one such hole, by using the ⅛ in. nominal diameter ceramic spheres of density 2·5 g cm^{-3}, indicates that the required full scale mass flow rate will be obtained by pitching the holes at the corners of equilateral triangles having 6½ in. sides. The stream of particles issuing from each hole is distributed laterally by impingement on a 60° cone located with its apex about ⅛ in. below the orifice.

While this solution is valid for the ⅛ in. diameter ceramic spheres, the particles produced in the full scale plant are unlikely to be truly spherical and will cover a range of sizes. Furthermore, their temperature in the distribution zone is likely to be of the order of 500 °C. Under these conditions, the seed particle surface could well be sticky and it is proposed to extend this study to include tests at the elevated temperature on the correct material. These will check the flow properties of the hot particles and will indicate whether larger holes and/or funnel-shaped entries to the holes are necessary.

2·3·5. *Aerodynamics*

Prediction of the flow behaviour and uniformity of particle dispersion through the two chambers is one of the most imponderable features of the design at the present time. Any analytical approach based on diffusion of particles in an isotropically turbulent system is of limited value as local temperature gradients are bound to exist and give rise to asymmetrical secondary flows. An empirical approach using models is therefore being adopted.

Preliminary theoretical and model studies have been made to determine the optimum inlet and outlet geometries to give uniform vertical flow in the upper chamber. As a broken line in Fig. 2.37 indicates the performance of the pump if the diode resistance ratio were 34 as in the case of the water model.

The outstanding problems now remaining before the fluid diode pump can be applied in the MHD field are: (i) the lack of suitable materials to withstand the highly corrosive slag, (ii) control problems—slag-level detecting device used to trigger the gas valves, (iii) realistic start-up and shut-down procedures. If solutions can be achieved in these crucial aspects the fluid diode pump would offer a practical answer to the problem of pumping liquid slag. This pump has no moving parts in contact with the slag, and although the efficiency would be inherently low it would be capable of high delivery pressures.

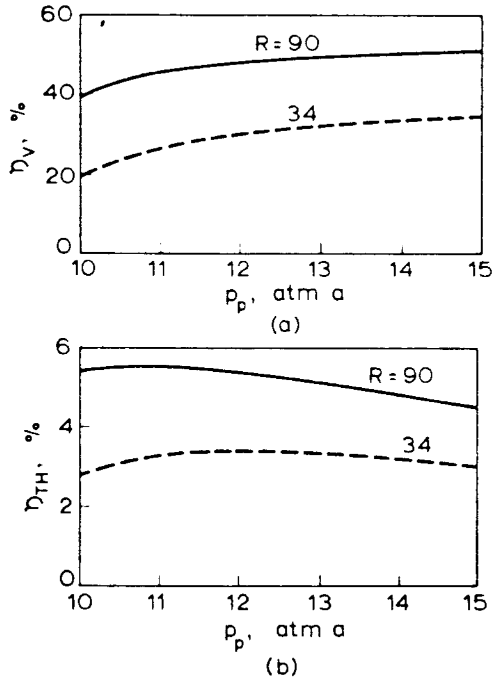


Fig. 2.37. Fluid diode—predicted slag pump performance. (a) Variation of volumetric efficiency with gas pumping pressure. (b) Variation of thermal efficiency with gas pumping pressure. Conditions: $p_2 = 9$ atm; $p_2 = 2$ atm; $p_s = 1$ atm.

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p. 79-85.

2.2.3b. *Reciprocating jet pump (vector flow)**

The reciprocating jet pump is based on the use of a non-reciprocal hydrodynamic effect (Walkden, 1967; Walkden and Kell, 1967). A conventional jet pump (Müller, 1964) is valveless and uses a fluid jet separately pumped into a suitably shaped main pipe, which carries the fluid to be pumped. The main body of fluid is propelled along this pipe by a momentum transfer and mixing process. The injected fluid must, however, be suitable for pumping by conventional means, be compatible with the pumped fluid, and be acceptable as a diluent. At high temperatures these conditions will usually eliminate all fluid combinations apart from a gas or vapour to pump a liquid, and vice versa. Unfortunately there is then a large mismatch in density which causes considerable energy loss during momentum mixing.

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* This section was written by A. J. Walkden; assistance was given by B. D. Eveleigh, J. B. Stutley, R. A. Chapman and R. L. Barley, the General Elcetric Company Limited.

The technique used in the reciprocating jet pump is to extract from the main pipe an appropriate amount of the fluid to be pumped, in readiness for re-injection. During this extraction or "suck" stroke the fluid mass removed at a sink or jet nozzle contains very little momentum prior to entering the nozzle. During the injection or "blow" stroke, however, the jet nozzle acts as a source of momentum directed as desired. This is then coupled to the surrounding fluid in the main pipe, with an unavoidable but not excessive energy loss. Over a complete "suck-blow" cycle a net forward thrust is created, which may be used to generate a pressure rise or a velocity increase in the main body of fluid, depending on the shape chosen for the main pipe. A similar principle has been studied for use in ship propulsion (Dickmann, 1950).

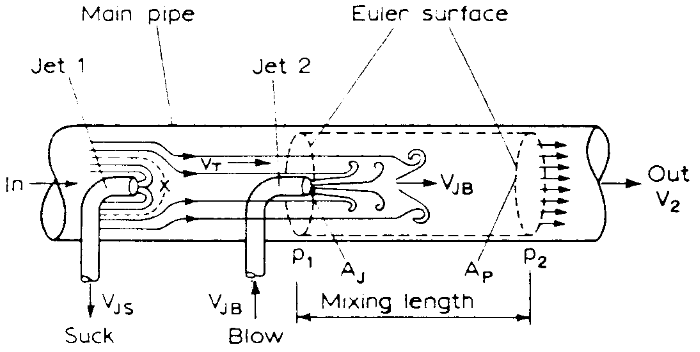

FIG. 2.38. Simple reciprocating jet pump showing suck and blow patterns.

The non-reciprocal action of an alternating jet depends upon the fact that the flow pattern for a steady blow stroke is fundamentally different from that for a steady suck stroke. This is shown in Fig. 2.38 and has been observed in a two-dimensional water tank using polystyrene beads as markers. The left-hand jet is sucking and the right-hand blowing. During the suck stroke irrotational potential flow occurs with a convergent "scalar" flow pattern. The flow pattern of the steady-state blow stroke might be regarded as a "vector" pattern.

The problem of pumping a steady stream of fluid has now been transformed to that of cycling fluid through a nozzle. In a severe environment the latter problem is much more simple to solve, for example by use of a vertically reciprocating gas–liquid interface stabilized by gravity. This forms a piston in a chamber coupled to the jet nozzle by means of a suitable diffuser. In the case of a pump for liquids, the gas pressure above the liquid surface may be cyclically varied by means of a reciprocating mechanical piston or bellows or by a valve-controlled supply of compressed gas, so as to blow and suck alternately through the jet orifice.

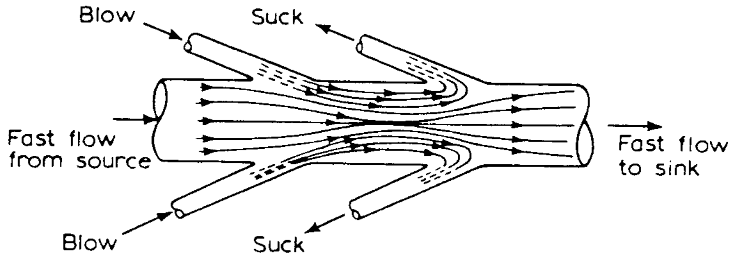

FIG. 2.39. Tree configuration of reciprocating jet pump for high velocities.

The simplest form of pump comprises a pipe with injector tubes or jet nozzles arranged at various points along the main pipe of the pump, and directed in the flow direction (Fig. 2.38). Several possible designs of pump have been proposed (Walkden and Kell, 1967), two of which are particularly useful. In the first practical design of reciprocating jet pump (Fig. 2.39) the jet tubes are fitted to the pipe like the branches of a tree, with no extension within the pipe. This design has the advantage of simple construction, low resistance to flow in the pipe, and least surface area open to corrosion by seed material. The design will be most useful when the requirement is for a high flow rate across a small pressure gradient and when efficiency is of little importance.

The second practical design is the "bulb-venturi" pump, shown in Fig. 2.40, which includes the air–liquid piston chambers. The pipe is shaped so as to provide a venturi and diffuser mixing chamber. The main purpose of the former is to accelerate the main liquid stream to a velocity nearly matching that of the jet so that less energy is lost during momentum mixing; the diffuser converts the velocity head back to pressure during the mixing. Each stage incorporates two jets cycling in anti-phase, so as to produce a more even thrust and continuity of mass. A larger number of suitably phased jets per stage would produce smoother flow, but space is limited close to the venturi. The jet nozzles have a rounded profile and a 10° diffuser angle so as to reduce losses and cavitation during the suck stroke. Streak photographs of two-dimensional flow patterns in a shallow tank have shown that, with the correct geometry, there is negligible interference between the blowing and sucking jets. This design will be most useful in a system requiring low flow rates across large pressure gradients.

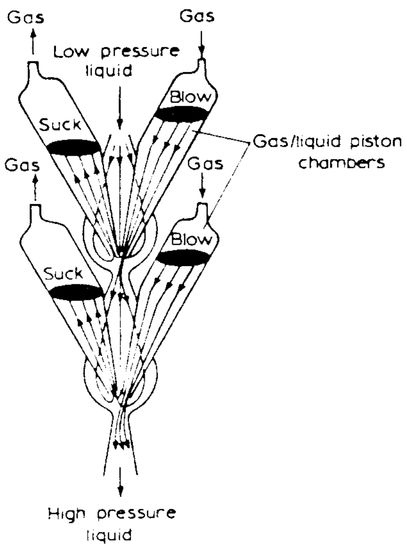

FIG. 2.40. Two-stage twin-jet bulb-venturi reciprocating jet pump.

A simple theory (Walkden and Kell, 1967) based on separate consideration of quasi-static blowing and sucking, has been found to give reasonable agreement with the results, providing boundary-layer corrections are applied. The pump head h may be expressed

$$h_P = nk\bar{v}_{JB}^{\,2}\,\alpha(1-\alpha/2-2v_P/\bar{v}_{JB})/g, \qquad (2.19)$$

where n is number of jets, k is the ratio of blow period $\bar{v}^2$ to the period of one cycle, $v_{JB}^{\,2}$ is the mean square jet velocity during T_B, α is the effective area ratio of jet nozzle to venturi or main pipe, v_P is the liquid velocity in the venturi or main pipe, $\bar{v}_{JB}$ is the mean jet velocity, g is acceleration due to gravity.

At high frequencies, or short wavelengths as defined by the ratio of pipe velocity to frequency, the steady-state treatment is likely to be less accurate. In order to allow for transients during switchover, a more sophisticated approach might be required which involves consideration of ring vortex streets (Dickmann, 1950).

Equation (2.19) relates pump performance to the jet velocity either as

measured, or as programmed by hydraulic coupling. However, it does not consider the way in which the jet velocity was achieved, for example by pneumatic coupling to a mechanical piston with a sinusoidal volume displacement. This has required a more extensive treatment since the drive pressure is related to the results it achieves, so the differential equations are non-linear and must be solved with the help of an analogue computer. If, however, the pressure variation as a function of time is known, either from a recorder or as a result of programming a valve-controlled compressed air supply, it is possible to estimate pump performance.

A simple mathematical model may be used to describe the behaviour of two piston chambers attached to nozzle/diffusers, which feed liquid alternately from one to the other via the pump body. Assuming gravitational and inertial effects within the piston chambers to be small, but taking into account typical nozzle/diffuser losses, the mean square of the jet blow velocity available from a twin-jet combination at the rth stage may be expressed

$$\bar{v}_{JBr}^{\,2} = 2C_J\overline{\Delta p_r}/\rho, \qquad (2.20)$$

where C_J is the nozzle/diffuser loss coefficient (= 0·67 for example), $\overline{\Delta p_r}$ is the mean of the modulus of the difference in pressure between the two rth piston chambers during an average cycle, ρ is the liquid density. Substituting eqn. (2.20) in eqn. (2.19) for each stage, with $k = \frac{1}{2}$, we have the pump stagnation head ($v_p = 0$) for n twin-jet stages given by

$$h_{PO} = 2C_J\,\alpha(1-\alpha/2)\left[\sum_1^n \overline{\Delta p_r}\right]\Big/\rho g. \qquad (2.21)$$

Experience has shown that the suck contribution to the pressure differential $\overline{\Delta p_r}$ sometimes has little effect, especially when cavitation occurs. Under these circumstances the use of the mean blow pressure alone in eqn. (2.21) gives satisfactory agreement with experiment.

A small, hot, single-stage twin-jet model, similar in principle to Fig. 2.40, but using piston chambers curved through 90°, has been constructed from silica glass with a jet diameter of 0·7 cm and venturi bore of 1 cm. This has pumped molten potassium sulphate at 1500°K at flow rates up to 180 $\text{cm}^3\,\text{s}^{-1}$ across a head of 13·5 cm. This corresponds to a stagnation head of 51 cm (6·8 cmHg). The pump was pneumatically coupled via a pair of water-cooled pipes to two mechanical pistons of 60 cm^3 displacement, which were sinusoidally cycled in anti-phase at frequencies in the range 2–5 Hz.

Water experiments have been carried out using a two-stage model closely similar to Fig. 2.40, with jet and venturi dimensions identical to the silica pump. This pump was driven by four servo-controlled spool-valves switching air pressures of up to 1·3 atm (100 cmHg). At this drive pressure, a stagnation head of 50 cmHg (8 cm for the first stage) was achieved at a frequency of 0·86 Hz, without the need for a sub-atmospheric suck stroke. This is because a small static pressure head at the first stage is sufficient to provide a reasonable jet suck velocity when assisted by the dynamic pressure rise in the nozzle/diffuser. A much greater head is produced by the second stage (42 cm), since the suck stroke is helped by the pressure developed by the first stage. The suck stroke of downstream stages is more effective due both to the increased pressure, and to the consequent alleviation of any limitation in suck velocity as a result of cavitation.

Considerable experimental work has been carried out using a pump of similar jet and venturi geometry, but with long straight-sided 1·6 cm bore piston chambers driven by the pneumatically coupled mechanical drive described earlier. A stagnation head of 32·5 mmHg has been achieved at 2·5 Hz for two stages each with a mean blow pressure of 0·45 atm (34·5 cmHg).

There are basically three efficiencies which require consideration with the reciprocating jet pump. Firstly, there is the hydrodynamic or pumping efficiency, the efficiency of conversion of liquid energy in the air–liquid piston chamber to work done on the pumped liquid. Secondly, there is the pneumatic efficiency, the efficiency of conversion of pneumatic energy to liquid energy in the piston chamber, and, finally, there is the efficiency of gas compression in creating the pneumatic energy.

The main drawback to conventional jet pumps is that the pumping or hydrodynamic efficiency is not more than 38%, mainly as a result of mixing losses. There are, however, two important differences when the reciprocating jet pump is compared with this. Firstly, there is a suck stroke, which consumes power while providing no output, but, secondly, there is much less restriction on the allowed values of the ratio of jet to venturi area. This ratio could approach unity with a reciprocating jet, which could result in a considerable increase in efficiency to near that for a diffuser (say 70%).

If a mechanical piston, bellows or alternative closed system is used, the pneumatic efficiency could be near 100%, but unfortunately the efficiency with which the pneumatic energy is created is likely to be low. Alternatively, compressed gas drive may be used which has been efficiently produced. However, it is likely that some pneumatic energy will be discarded to the exhaust on the suck stroke.

In a hot system the liquid enthalpy could perhaps provide some power by expanding injected cold gas or generating steam, but these possibilities remain to be investigated. It is difficult to generalize, but the overall efficiency is unlikely to be greater than 40%.

2.2.3c. *Air lift pump**

The air lift pump (Nicklin, 1963; Perry, 1953) relies on a hydrostatic balance between a column of liquid and an aerated column of the same liquid, known as the riser, in which the aeration reduces the average density and hence increases the liquid height. The difference in these heights is the pump stagnation head, which is maintained by compressed air injected at the bottom of the aerated column.

In the discussion below, flow resistance effects are neglected, and the theoretical efficiency is derived from a consideration of point conditions in a two-phase system, integrated over a column. Figure 2.41 shows a typical configuration with submergence depth h_s of the compressed air source below the lower liquid level, lift height or pump stagnation head h_p, and riser height h_r. Simple hydrostatic considerations show that if α is the volume fraction occupied by liquid in the riser, then

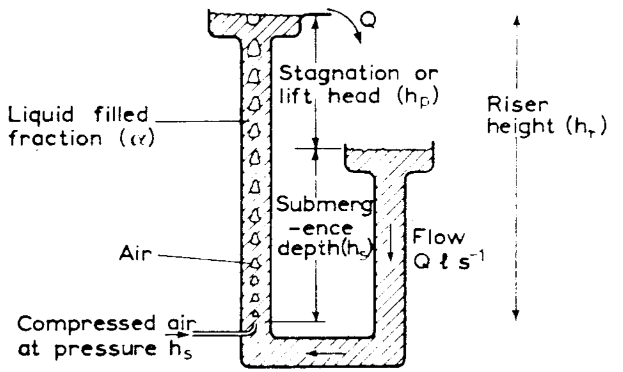
Q
Stagnation or lift head (h_p)
Riser height (h_r)
Liquid filled fraction (α)
Submerg -ence depth(h_s)
Flow Q ℓ s^{-1}
Air
Compressed air at pressure h_s

FIG. 2.41. Air lift pump.

$$h_s = \alpha(h_p + h_s) = \alpha h_r \quad \text{and} \quad h_p = (1 - 1/\alpha)\, h_s;$$

also

$$h_r = h_p/(1-\alpha).$$

The various heights of an air lift pump for equilibrium seed may hence be calculated using the fact that for equilibrium seed at 1700°K, h_p must be at least 6 m atm^{-1} of pump head required.

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* This section was written by A. J. Walkden, the General Electric Company Limited.

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Reciprocating-jet Pump

FOR HOT CORROSIVE FLUIDS

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R. C. Kell joined the staff of G.E.C. Research Laboratories in 1948 after graduating in Physics at Bristol University. He worked on synthetic quartz until 1954, and was responsible for research and development on ferroelectric ceramics and crystals (1954-7) and on magnetic films for computer memories (1957-64). Currently he is responsible for the M.H.D. and Materials Research Groups.

When pumping hot corrosive fluids it is desirable (and sometimes essential) to avoid the use of solid moving parts in contact with the fluid. In some applications an injector pump[1] can be used, but the drive fluid, e.g. air, water or the vapour of the liquid to be pumped, must then be suitable for pumping by conventional means. The use of two fluids often has disadvantages and the present pump has been devised to be driven with the same fluid as that being pumped.

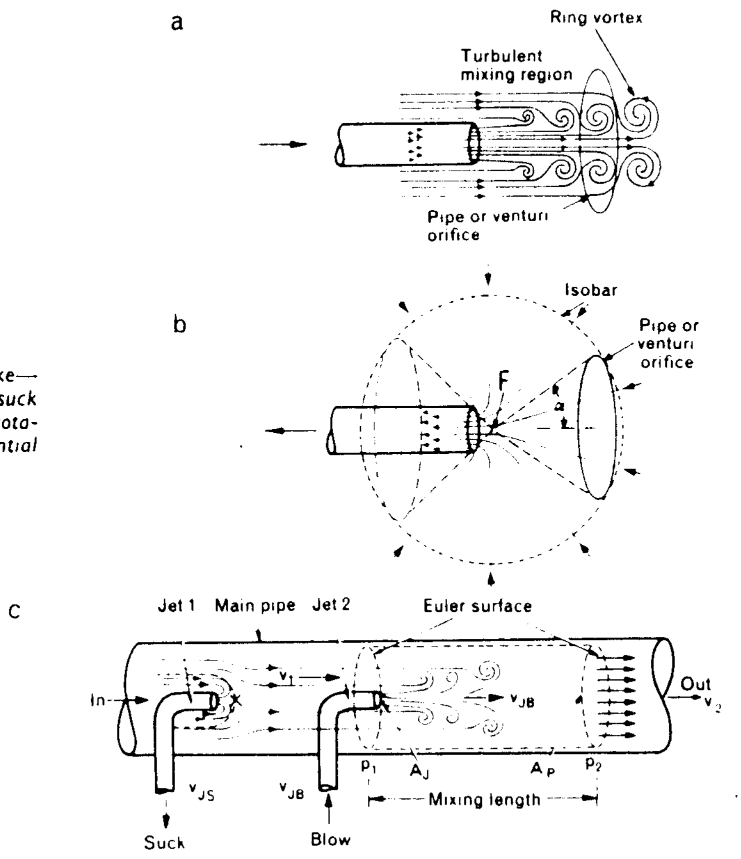
PRINCIPLES OF OPERATION

Operation of the reciprocating-jet pump depends on the basic difference between the flow pattern produced by steady blowing of a fluid through an orifice into a fluid-filled pipe and that produced by steady sucking through the orifice. The pump comprises a modified form[2] of injector pump in which fluid is alternately blown out of a jet orifice into a main pipe and then sucked back into the orifice. This can be achieved either by hydraulic coupling to a cooler mechanical piston via immiscible liquids, or preferably by pneumatic coupling in which the air liquid interface acts as a piston.

During the blow stroke, fluid is emerging at the jet orifice, or source, which determines the direction of its momentum. Provided there are no pressures or surfaces able to redirect this momentum, the flow continues until turbulently mixed with and viscously damped by the surrounding liquid in the main pipe. A 'vector' flow pattern results, as in fig. 1a. If required the injected momentum may be converted into a pressure rise by means of a diffuser.

During the suck stroke the orifice acts as a sink which extracts the same mass of fluid as had been injected during the previous blow stroke. The flow pattern and momentum properties of material just before removal at the sink orifice are dictated by the relative flow rates of fluid entering and fluid passing by the orifice, and by the transient effect of the flow pattern just before commencement of the suck stroke. Only these boundary conditions, plus a few shape factors, have any marked influence on the directional distribution of momentum extracted from the body of the liquid surrounding the orifice. The direction in which liquid moves once it has entered the sink has no influence on the external momentum.

Two extreme cases may be cited as examples of two possible suck patterns. Firstly, consider the



1 *Reciprocating-jet pump:* ***a*** *blow stroke—'vector' flow pattern (turbulent);* ***b*** *suck stroke—'potential' flow pattern (irrotational);* ***c*** *simple design showing potential flow patterns*

convergent potential-flow pattern which occurs when sucking through an orifice at a high pressure difference from an initially stationary infinite volume of fluid. This might be described as a 'scalar' flow pattern. Isobars are in the form of spheres concentric with the sink, and fluid flows radially towards the focal point *F* formed by the sink, as in fig. 1b. In this case the net momentum is zero when integrated over a large sphere surrounding the orifice. In the pump, the corresponding situation occurs when the mass flow rate into the sucking orifice is very large compared with that in the main pipe. Approximately equal quantities of liquid then flow into the orifice from both directions.

The second extreme case occurs when sucking at infinitesimal pressure difference from a fast-moving fluid stream. Consider a short tube open at both ends placed at some angle, say 45° to the flow direction. Here the suck pattern and resolved momentum about the point of entry are dictated solely by the boundary conditions given by stream velocity, tube shape factors and Reynolds number. In the pump, the corresponding situation occurs when the flow rate into the sucking orifice is very small compared with that of the fast-moving fluid in the main pipe. The sucking orifice then has negligible effect on the flow pattern.

In the intermediate and more practical case shown for the first jet in fig. 1c, the flow rate into the sucking orifice is less than, but of a similar order of magnitude to, that in the main pipe. The liquid flowing along the pipe towards the orifice then consists of two concentric streams separated by a boundary surface of stream lines. One stream flows into the jet orifice from behind it and the other flows past the orifice and diffuses until the velocity across the pipe is again uniform. The large transverse velocity gradient produced across the boundary between the two streams adds a small driving force to the passing liquid.

In the reciprocating-jet pump, fluid is alternately blown out of the orifice and sucked into it. The blow stroke then produces entrainment, and therefore drives the fluid along the pipe in the required direction, while the suck stroke produces at worst no net reversing force on the fluid in the pipe. The non-reciprocal behaviour is associated with the energy losses, during the blow stroke, due both to turbulent mixing with production of vortices and to viscous damping.

POSSIBLE DRIVE SYSTEMS

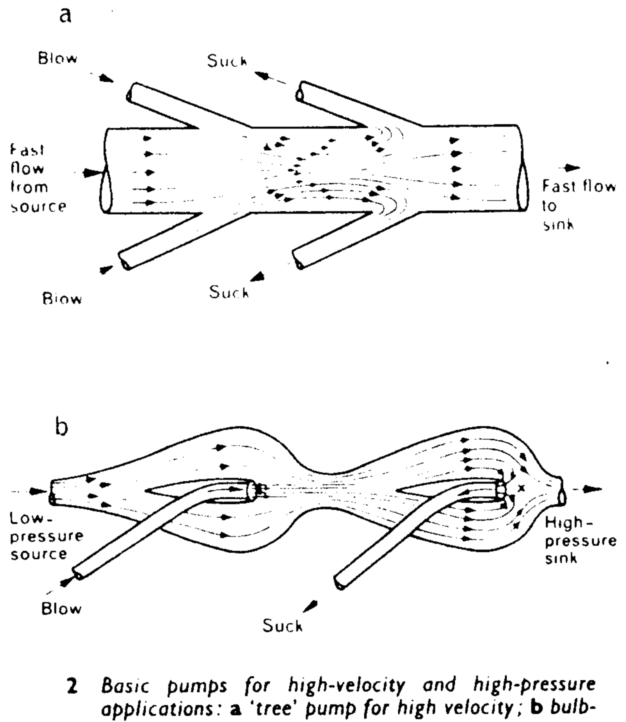
The problem of pumping a fluid without the use of solid moving parts in contact with the fluid has now been exchanged at some loss of efficiency for that of cycling fluid into and out of a jet orifice. The methods discussed below are concerned with the pumping of liquids, but they may be adapted for pumping gases, vapours and solid suspensions. Cycling of the jet liquid can be achieved by means of hydraulic coupling to a mechanically driven solid piston via compatible immiscible liquids if such suitable liquids exist. Alternatively, the jet liquid can be driven by a gas whose pressure is varied—for example by pneumatic coupling, either to a mechanically driven piston or to a valve switched alternately to high- and low-pressure gas chambers. The piston/gas drive is the more simple and efficient method, but high gas temperatures may involve difficult problems. The valve/gas drive is less easy to control and is likely to be less efficient, since hot, partly compressed air is discarded. Preferably, the gas/liquid interface should be in a 'piston-chamber' larger in diameter than the jet, to reduce its movement.

A fourth method involves the production of pulsating pressure by combustion of fuel in air within the gas/liquid piston chamber in the manner of a two-stroke internal combustion engine. However, control may be difficult.

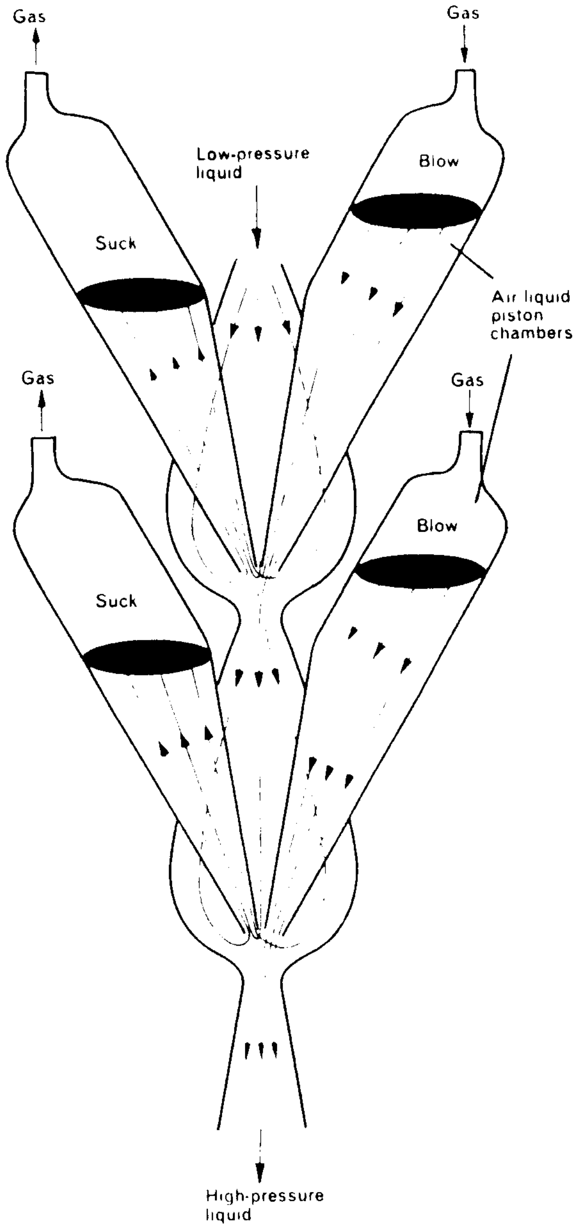
Arising from this is a more speculative method which may be used if the liquid being pumped is very hot. Water or some other suitable liquid may be injected below the surface of the hot liquid within the gas/liquid piston chamber and vapourized. An increase in pressure would result at the expense of a small temperature drop in the pumped liquid and the liquid would be driven downwards out of the chamber. The chamber would then be opened to a condenser for the suck stroke. This system has the advantage of great simplicity and efficiency, since no separate fuel or compressors are required. The disadvantage again is that control may be difficult.

SOME POSSIBLE DESIGNS

The simplest form of pump comprises a pipe with injector tubes or jets arranged at various points along the pump and directed in the flow direction (fig. 1c). The pipe and jets are filled with liquid. Suitably phased alternating pressure pulses are applied to the jets by one of the methods just described. During the blow stroke, the pressure in the jet is greater than in the pipe so that liquid flows out of the jets and produces 'vector' flow along the pipe. During the suck stroke, the pressure is reversed and an equal amount of liquid flows back into the jet to produce a convergent or 'scalar' flow pattern.



2 *Basic pumps for high-velocity and high-pressure applications:* ***a*** *'tree' pump for high velocity;* ***b*** *bulb-venturi pump for high pressure*



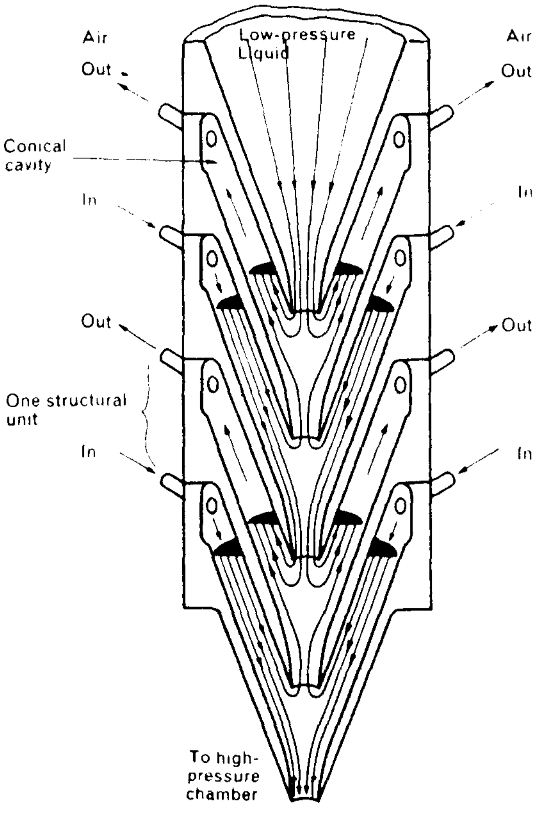
3 *Two-stage twin-jet bulb-venturi pump*

In the first practical design of reciprocating-jet pump (fig. 2a) the jets are fitted to the pipe like the branches of a tree, with no extension within the pipe. This design has the advantage of least resistance to flow in the pipe and also least surface area open to corrosion by the liquid being pumped. The design will be most useful when the requirement is a high flow rate across a small pressure gradient.

The second practical design is the 'bulb-venturi' pump (fig. 2b). The pipe is shaped so as to provide a venturi and diffuser. The main purpose of the former is to accelerate the main liquid stream to a velocity nearly matching that of the jet so that less energy is lost during momentum mixing; the diffuser converts the velocity head back to pressure during the mixing. This design will be most useful in a system requiring low flow rates across large pressure gradients.

The third design (fig. 3) utilizes the 'bulb-venturi' shape, but each stage incorporates two jets cycling in anti-phase, so as to produce a more even thrust. A larger number of suitably phased jets per stage would produce smoother flow, but space is limited close to the venturi. The air/liquid piston chambers are also shown in fig. 3. Streak photographs[2] of two dimensional flow patterns in a shallow tank have shown that, with the correct geometry, there is negligible interference between the blowing and sucking jets.

A fourth design, which might be considered a topological distortion of fig. 2a, consists basically of a series of similar inverted cones which are concentric, with a vertical axis as shown in fig. 4. Each conical cavity thus formed is sealed at the top, with small air pipe connexions, and has a shaped orifice at the bottom directed into the cavity beneath. Initially the top half of each cavity is filled with air and the lower half with liquid which is continuous throughout all cavities. During the first half of a pump cycle, a high air pressure is applied to alternate cavities and a low pressure to the intermediate cavities. Liquid is thus forced into the low-pressure cavities. Owing to the shaping of the orifice and to the creation and ejection of momentum downwards from the high pressure cavity, liquid is forced preferentially into the cavity below. The second half of the pump cycle involves the reversal of the whole operation in a given cavity, although for the pump as a whole there is a net transfer of liquid downwards against the pressure gradient. This design is particularly suitable for viscous liquids.



4 *'Cone' pump for viscous liquids*

THEORETICAL ANALYSIS

The fundamental difference between a continuous suck stroke and a fully developed blow stroke has been described above. A hydrodynamic analysis of the development of the two flow patterns is very difficult and will not be dealt with here. Each stroke is therefore given a separate mathematical treatment by means of suitable steady-state models, before consideration of the complete cycle in which there is zero net addition of liquid. The two strokes will be analysed for a single-jet pump in terms of the jet force F_J, or equivalent pump head h_P. The jet force will be given by the sum of the pressure force developed and the rate of change of momentum of the liquid which flows past the jet while a steady jet velocity of either v_{JB} on the blow stroke, or $-v_{JS}$ on the suck stroke, is applied.

Blow Stroke

For the blow stroke, use is made of the generalized form of the Euler momentum theorem, which may be stated[3] as follows:

The resultant thrust on a fluid contained in a closed surface S is equal to the reversed resultant of the body forces on the enclosed fluid, together with the rate of change of momentum of the fluid and the rate of flow of momentum outwards across the boundary of S.

The following assumptions are made:

(i) the body force (e.g. gravity) is zero,
(ii) steady-state conditions have been achieved,
(iii) the density ρ is constant, and
(iv) the jet area A_J is small compared with the pipe area A_P.

The Euler Equation may now be written:

$$(p_2-p_1)A_P = \rho A_P v_1^2+\rho A_J v_{JB}^2-\rho A_P v_2^2, \dots .(1)$$

where p_1, p_2; v_1, v_2 are the inlet and outlet pressures and velocities, respectively.

The Continuity Equation gives:—

$$\rho A_J v_{JB} = \rho A_P(v_2-v_1) \quad \dots\dots\dots(2)$$

The force F_{JB} developed by the pump during a steady blow stroke is given by the pressure force plus the rate of increase in momentum from the input.

$$F_{JB} = A_P(p_2-p_1)+\rho A_P v_1(v_2-v_1) \quad \dots\dots(3)$$

Substituting equations 1 and 2 in equation 3, and using $\alpha = A_J/A_P \ll 1$,

$$F_{JB} = \rho A_J v_{JB}^2\left(1-\frac{\alpha}{2}-\frac{\bar{v}_P}{v_{JB}}\right) \quad \dots\dots\dots(4)$$

where the mean pipe velocity is $\bar{v}_P = \frac{1}{2}(v_1+v_2)$.

Suck Stroke

The suck stroke may be described roughly by the hydrodynamics of an ideal incompressible, inviscid, irrotational liquid. An ideal liquid passing the sucking orifice is separated from that entering the orifice by a set of streamlines which form a boundary surface as shown in fig. 1c. In general the streamlines will describe the mean velocity vector of turbules. The boundary streamlines combined with the pipe wall will form a Bernoulli tube of flow; hence on the assumption of a perfect diffuser,

Bernoulli's Theorem gives:—

$$\frac{p_2 - p_1}{\rho} = \tfrac{1}{2}(v_1^2 - v_2^2) = \bar{v}_P(v_1 - v_2). \quad \ldots.(5)$$

The Continuity Equation gives:—

$$(v_1 - v_2) = \alpha v_{JS}. \quad \ldots\ldots\ldots.(6)$$

The force F_{JS} developed by the pump during a steady suck stroke is given by the pressure force plus the rate of change in momentum of the liquid in the pipe.

$$F_{JS} = A_P(p_2 - p_1) + \rho A_P(v_2^2 - v_1^2). \quad \ldots.(7)$$

Substituting equations 5 and 6 in equation 7

$$F_{JS} = -\rho A_J \bar{v}_P v_{JS}. \quad \ldots\ldots\ldots.(8)$$

This treatment takes no account of any 'turbulent pull effect' caused by the diffusion of momentum by means of break-away turbules. These turbules will transfer momentum from the fast-moving liquid flowing into the sucking jet to the liquid passing the jet. This extra momentum will tend to cancel any losses in the diffuser of equation 5, so that the assumption of a perfect diffuser is probably realistic.

When equations 4 and 8 are compared it may be seen that when $\bar{v}_P$ is small and $v_{JB} \simeq v_{JS}$ then $F_{JS} \simeq -F_{JB}\bar{v}_P/v_{JB}$. Hence the blow stroke has a much larger effect than the suck stroke, as expected.

Blow-suck Cycles

Let I_P be the total impulse given by the pump per cycle; we may then obtain F_P, the pump force averaged over one cycle, defined as:

$$F_P = \frac{I_P}{T} = \frac{1}{T}\int_0^{T_B} F_{JB}\,dt + \frac{1}{T}\int_{T_B}^{T} F_{JS}\,dt, \quad \ldots.(9)$$

where $T = T_B + T_S$, the period of one complete cycle,

T_B is the blow period

T_S is the suck period

Equation 9 may be simplified to

$$F_P = k F_{JB} + (1 - k)F_{JS}, \quad \ldots\ldots.(10)$$

where $k = T_B/T$

F_{JB}, F_{JS} are the average forces during the blow and suck periods, respectively.

The condition for zero net mass transfer during a cycle is given by:

$$\bar{v}_{JS} = \bar{v}_{JB}\frac{k}{1-k} \quad \ldots\ldots\ldots\ldots(11)$$

where $\bar{v}_{JS}$, $\bar{v}_{JB}$ are the mean jet velocities during the suck and blow strokes, respectively.

The pump force for a system having n jets is n times that for a single jet, given by substituting equations 4, 8 and 11 in equation 10. Thus:

$$F_P = nk\rho A_J V_{JB}^2\left(1 - \frac{\alpha}{2} - \frac{2\bar{v}_P}{\bar{v}_{JB}}\right) \quad \ldots\ldots(12)$$

where V_{JB} is the root-mean-square jet blow velocity.

Pumps are usually described in terms of the head developed, which is given by

$$h_P = \frac{F_P}{A_P\rho g} = nk\alpha\frac{V_{JB}^2}{g}\left(1 - \frac{\alpha}{2} - \frac{2\bar{v}_P}{\bar{v}_{JB}}\right) \quad \ldots.(13)$$

where g is the gravitational acceleration.

Pump Characteristics in a System

When a pump is used in a system, it must produce the required increase in total head. The particular combination of pressure, gravity and velocity heads may be achieved by using a suitable inlet nozzle and outlet diffuser. Once steady flow in the pipe has been achieved, the pump operates against a system head (h_S) comprising the total useful head (h_A) added to the pumped fluid, the loss head (h_L) due to energy losses in any conversion between potential and kinetic energy, and the resistance head (h_R) due to frictional losses at solid/liquid surfaces. Thus:

$$h_S = h_A + h_L + h_R \quad \ldots\ldots\ldots.(14)$$

In this equation,

$$h_A = h_0 + p_0/\rho g + (v_{out}^2 - v_{in}^2)/2g$$
$$h_L = (S_{in} + S_{out})V_P^2/g$$

and for Reynolds numbers over 3000,

$$h_R = R_1 V_P^2/g$$

where h_0 is the difference between output and input gravity heads

p_0 is the difference between output and input pressure heads

v_{in}, v_{out} are the inlet and outlet velocities of the system respectively

S_{in}, S_{out} are the shape-dependent loss factors of the inlet nozzle and outlet diffuser, respectively, where

$0 < S_{in} < \tfrac{1}{2}$ and $0 < S_{out} < \tfrac{1}{2}$.

V_P is the root-mean-square pipe velocity

R_1 is the specific surface resistance × wetted area: this term decreases slightly with increasing Reynolds number.

In the steady state, the flow will be self-adjusting so that the system head becomes equal to the pump head. Hence the steady state equation of motion for Reynolds numbers over 3000 is:

$$k\alpha\frac{V_{JB}^2}{g}\left(1 - \frac{\alpha}{2} - \frac{2\bar{v}_P}{\bar{v}_{JB}}\right) = h_0 + \frac{p_0}{\rho g} + \frac{v_2^2 - v_1^2}{2g} + R\frac{V_P^2}{g} \quad (15)$$

where $R = R_1 + S_{in} + S_{out}$

If the pump operates between two large tanks exposed to the atmosphere, h_0 represents the vertical difference between the two liquid surface levels, $p_0 = 0$ and $v_1 = v_2 \simeq 0$. If also α and $\bar{v}_B/\bar{v}_{JP}$ are both less than 0·1, equation 15 simplifies to:

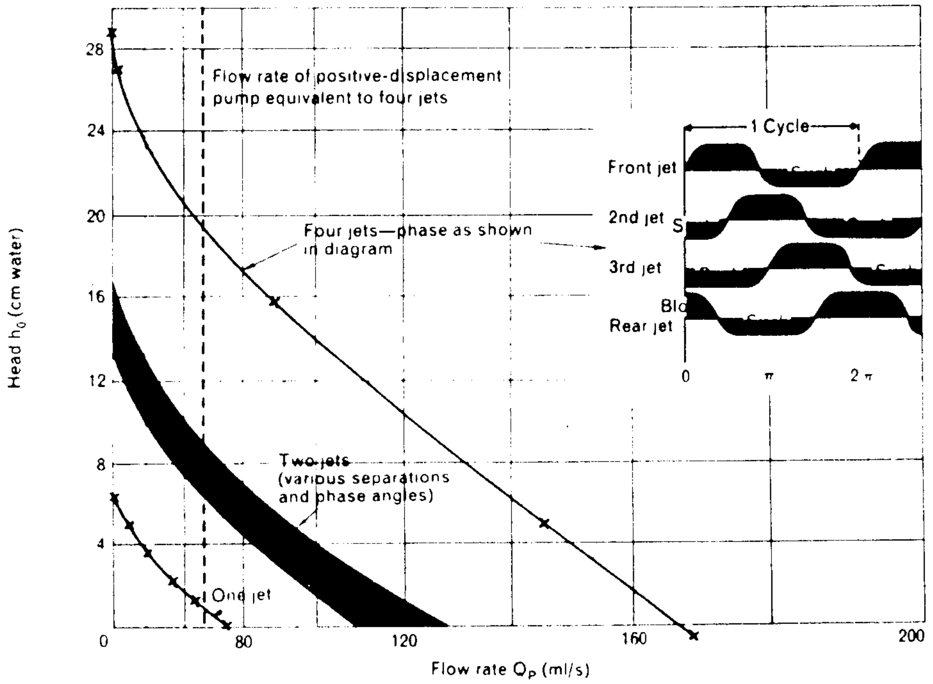
$$V_P^2 \simeq \frac{k\alpha}{R}V_{JB}^2 - \frac{g}{R}h_0. \quad \ldots\ldots.(16)$$

For any given pump geometry and cycle waveform, the parameters k, α, and R are fixed. Equation 16 then defines a plane if V_P^2, h_0 and V_{JB}^2 are plotted along the X, Y and Z axes respectively. The intercepts of this characteristic plane on the co-ordinate planes give the following straight lines:

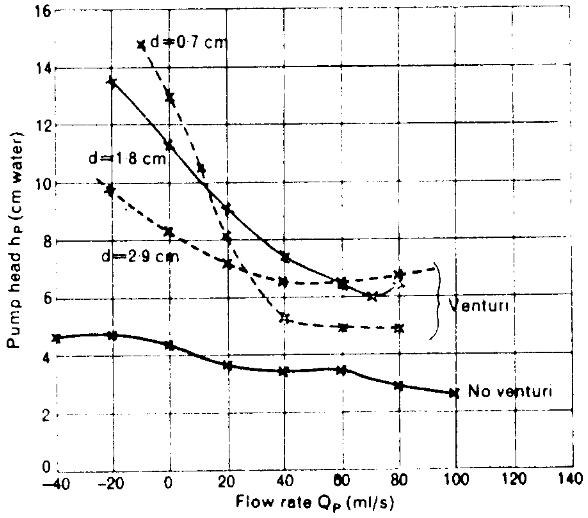
(i) Resistance line (XY plane, $V_{JB} = 0$),
(ii) Stagnation head line (YZ plane, $V_P = 0$),
(iii) Zero static head line (ZX plane, $h_0 = 0$).

APPARATUS

Initially it is advantageous to use a cold, non-corrosive liquid at a Reynolds number similar to that likely to arise in the final device. Either water or an oil will usually be suitable for this purpose. The pump performance can then be evaluated and the design developed using glass models in which



5 Characteristics of simple pump (fig. 1c) operated with one, two and four jets ($V_{JB}=4{\cdot}5$ m/s, $f=2$ c/s, $k=0{\cdot}4$, jet bore $=0{\cdot}35$ cm, pipe bore $=1{\cdot}5$ cm)



6 Effect of venturi on pump characteristics (single jet, $V_{JB}=3{\cdot}6$ m/s, $f=2{\cdot}0$ c/s, $k=0{\cdot}4$, jet bore $=0{\cdot}35$ cm, pipe bore $=1{\cdot}5$ cm, venturi bore $=0{\cdot}73$ cm, distance d between jet and venturi orifices is as shown on curves)

the liquid in the jets is reciprocated by direct hydraulic drive from mechanical pistons. If pneumatic drive is used, the compressibility of the gas complicates the evaluation of the pump and makes it more difficult to calculate the expected performance, since two coupled dynamic systems are involved.

The results described below were obtained by using water in a simple design of glass pump similar to that shown in fig. 1c. The main pipe was 20 cm long with a 1·5 cm bore and contained four jets of 0·35 cm bore spaced in series 3 cm apart. The pump was mounted vertically with each end connected by polythene tubing of 1·9 cm bore to constant-level water tanks whose vertical positions could be adjusted. The pressure head across the pump was thus known and the water flow rate could be measured. The mechanical drive incorporated four cam-driven pistons with variable stroke, frequency, relative phase and waveform. From these parameters the amplitude and waveform of the jet velocity could be derived. The cams were designed to give an approximately square waveform with the root-mean-square blow velocity (V_{JB}) about 1·6 times the root-mean-square suck velocity (V_{JS}) and k about 0·4.

RESULTS

Pump Head

The curve of pump head as a function of flow rate (or pipe velocity) for a given jet-velocity cycle is obtained from the characteristic and resistance curves. The characteristic curve (equation 16 with V_{JB}^2 constant) relates the flow rate to the static head for a given jet velocity. The resistance curve (equation 16 with h_0 negative and V_{JB} 0) relates the flow rate to the applied (negative) head without the jets operating. Each point on the pump-head curve is obtained by subtraction of the 'head' co-ordinate at the required flow rate on the resistance curve from the corresponding 'head' on the characteristic curve. In practice, turbulence may cause difficulty in obtaining the exact resistance curve corresponding to the conditions present when the characteristic curve was obtained.

Experimental results show that for $\bar{v}_P/\bar{v}_{JB}<0{\cdot}1$, the pump head is relatively independent of the flow rate but approximately proportional to V_{JB}^2 for given waveform, as expected.

Multiple Jets

Various combinations of two jets differing in relative phase and separation were measured at a constant jet velocity. Fig. 5 shows the results when h_0 is plotted against the flow rate Q_P squared. The curvature at low flow rates arises from the change from a square-law to a linear-law dependence of the resistance head on the velocity as the flow becomes laminar. The stagnation head is, in fact, larger than would be expected from measurements made at high flow rates. The two-jet results lie within a band giving approximately twice the performance of the single-jet curve. The points within the band show some values for the front jet leading in phase by $\pi/2$. Since this phase difference was the optimum for jets close together it was chosen for four-jet operation. The curve for four jets is also shown in fig. 5. Again there is a factor of two in performance compared with two jets, as expected.

Fig. 5 also shows the characteristics of a single-acting positive-displacement pump cycling the same volume as the four-jet system. The reciprocating-jet pump gives the better performance except at high pressure, since momentum is not destroyed by the closing of valves.

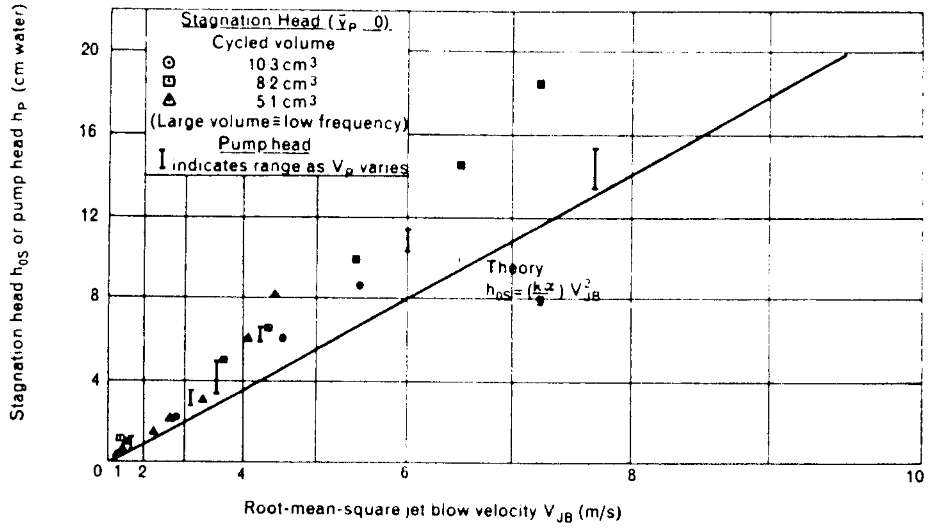
Effect of a Venturi

Fig. 6 shows the effect on the pump head of inserting a venturi of 0·73 cm bore in various positions. It gives a significant improvement at the expense of a higher internal resistance. The venturi position relative to the divergent blow stream has a marked effect on the mixing properties (and hence the pump head). The optimum position will depend on the required ratio of velocity head to pressure head.

Comparison with Theory

As described above, the intercepts of the characteristic plane (equation 16) on the co-ordinate planes give the resistance line, stagnation-head line and zero-static-head line, respectively. The resistance line (zero jet velocity) can be obtained only by experiment, owing to the different effects of laminar flow ($h_0 \propto V_P$) and turbulent flow ($h_0 \propto V_P^2$), and to the unknown resistance of the pipe, jets and connecting tubes.

The stagnation-head line (zero pipe velocity) is compared with experimental results in fig. 7. Ranges of values of pump head obtained as the pipe velocity is varied are also shown; these, as well as the stagnation-head values, lie above the theoretical line. This is consistent with the probability that the effective jet area is smaller than that assumed, owing to boundary-layer effects. The value of boundary-layer thickness which brings the experimental points down to the stagnation-head line is consistent with expected values. The zero-static-head line and the pump characteristic curves (constant V_{JB}^2) also show somewhat higher values than those predicted.



7 *Comparison of experiment with theory: variation of stagnation and pump heads with jet velocity*

EFFICIENCY

No attempt has yet been made to measure the efficiency of conversion of piston or gas energy into potential and kinetic energy of the pumped fluid. Curves[4] have, however, been published for the conventional injector pump giving efficiency of conversion of jet energy into energy of the pumped fluid as a function of the relative values of the input (H_1), output (H_2) and jet (H_J) total heads, with the ratio α of jet area to venturi area as a parameter. The maximum efficiency obtained is 38%, the optimum conditions being $\alpha = 0{\cdot}28$ and $(H_J - H_1)/(H_2 - H_1) = \gamma = 3$. If γ varies over the range $+\gamma_0$ to $-\gamma_0$ in a square waveform cycle having equal blow and suck periods which are sufficiently long for transient effects to be negligible, the maximum efficiency lies between 19 and 38%, the optimum conditions being as above. The lowest efficiency applies if during the suck stroke the input work equals that during the blow stroke and the output work is zero. The highest figure applies if during the suck stroke the input work is negligible or if the same proportion of it is converted into useful work as during the blow stroke.

If γ varies sinusoidally over the range $\pm\gamma_0$ and it is assumed that the instantaneous efficiency equals the steady-state value under the same conditions, the calculated maximum efficiency lies between 12 and 24% depending on the suck stroke as above, the optimum conditions being $\alpha = 0{\cdot}25$ and $\gamma_0 = 8$.

An estimate has also been made of the effect of a 50% sinusoidal variation in $H_2 - H_1$ when $H_J - H_1$ is sinusoidal. Such a large variation is unlikely to occur in practice. Under the worst possible phase conditions, which could apply to only a small proportion of the jets in a multi-jet system, the maximum efficiency lies between 7 and 14%, the optimum conditions being $\alpha = 0{\cdot}25$, $\gamma_0 = 13$.

APPLICATIONS

The main features of the reciprocating-jet pump are:

(i) The solid moving parts (pistons or valves) operating the jets need not be in contact with the pumped fluid and need not therefore be subject to corrosion, erosion or high temperatures, in contrast to the conditions in rotary and displacement pumps.

(ii) The driving fluid is the same as the pumped fluid, so that no dilution, aeration, temperature change or other contamination of the pumped fluid occurs, whereas the conventional injector pumps would require a second fluid if feature (i) were to be retained.

(iii) The dimensional tolerances are wide and the pump will still operate, though at reduced efficiency, when eroded or damaged.

(iv) The efficiency is relatively low (10 to 20%).

Applications in which isolation from the solid moving parts is advantageous include pumping hot and/or corrosive liquids (e.g. molten metals, glasses or salts) and liquefied gases (e.g. helium). Those in which absence of contamination is important include pumping blood or high-purity liquids. A further application is the pumping of solid suspensions in which large particles might otherwise give trouble.

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EXPERIMENTAL AND THEORETICAL STUDY OF A RECIPROCATING-JET PUMP USING PNEUMO-HYDRAULIC DRIVE

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SYNOPSIS

Experimental and theoretical studies of the performance of a reciprocating-jet pump are described. The pump has no solid moving parts in contact with the pumped liquid, and each jet is coupled pneumatically *via* a gas—liquid piston to a mechanically-driven solid piston. Experimental results and theory show good agreement when boundary-layer effects are taken into account. Analogue computer solutions of the equations of motion for the drive system are presented: these may be adapted to calculate the performance of other designs.

Introduction

The reciprocating-jet pump[1-4] was devised for use at very high temperatures with corrosive liquids such as the molten salts at over 1600°K which might be encountered in a magnetohydrodynamic generator.[5-7] An essential feature of the pump was to be the effective absence of solid moving parts, such as pistons, valves, and rotors so that it could be easily constructed from refractory oxides and remain relatively insensitive to erosion.

The operation of the pump depends on the basic difference between the flow pattern[1] produced when fluid is blown steadily out through an orifice into a fluid-filled pipe and that resulting from a steady sucking of liquid in through the orifice. Thus in effect a non-reciprocal hydrodynamic effect is used in association with the main principle of the conventional jet pump[8] which is the injection of momentum into the primary fluid by means of a second fluid. The novel feature of the reciprocating-jet pump is that the liquid to be injected is extracted from the liquid being pumped.

The "suck" stroke involves the extraction of an appropriate amount of liquid from the main body of fluid in readiness for re-injection during the "blow" stroke. Fluid emerging from the jet orifice during the blow stroke has a vectored momentum which is then coupled to that of the surrounding fluid with an irreversible but not excessive loss of energy. During the suck stroke the same mass of fluid is extracted as had been injected during the blow stroke but the directional distribution of momentum removed from the main body of fluid is determined only by the flow pattern near the orifice.

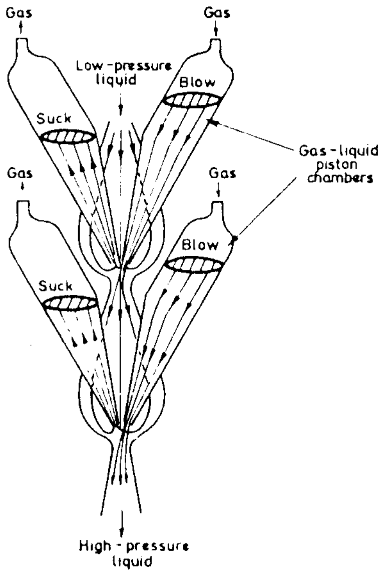

Fig. 1.—*Diagrammatic view of a two-stage twin-jet bulb-Venturi pump*

A number of pump designs were proposed and tested[2] and led to the design and construction of a glass model of the twin-jet bulb-Venturi version of the reciprocating-jet pump. The latter is the subject of this paper and a diagrammatic view is shown in Fig. 1.

By comparing experimental results with theoretical prediction it was hoped to approach a working theory which, if not completely describing the rather complex hydrodynamics of the pump, would at least predict the performance of the pump under various operating conditions.

Description of Pump

The simplest form of pump comprises a cylindrical pipe with injector tubes or jets arranged at various points along the pump and directed in the direction of flow. The pipe and jets are filled with liquid and suitably phased pressure pulses are applied to the jets so that liquid is cycled through the jet orifices and a pumping head is developed. Possible methods of applying the pressure pulses are described elsewhere.[2] Of these, the most simple approach is to hydraulically couple the jets to gas chambers in which the pressure is pulsated.

The pump studied in this paper is regarded as a practical hydrodynamic design and utilises the "bulb-Venturi" shape with two jets cycling in anti-phase so as to generate a uniform thrust. The main pipe is designed to accelerate the main stream of liquid through each Venturi to a velocity close to that of the blowing jet thus enabling momentum-mixing to occur with a minimum loss of energy. The diffuser converts the velocity-head back into pressure-head during the momentum-mixing process.

Injection and extraction of the liquid are achieved by a pneumo-hydraulic drive system which comprises a reciprocating mechanical piston coupled pneumatically to each "gas-liquid piston" chamber. The gas-liquid piston is defined by the common gas-liquid interface. The gas-liquid piston chambers in the experimental model used air as the drive gas and were made straight-walled and relatively long to ease the measurement of liquid strokes and tube velocities. Although wall

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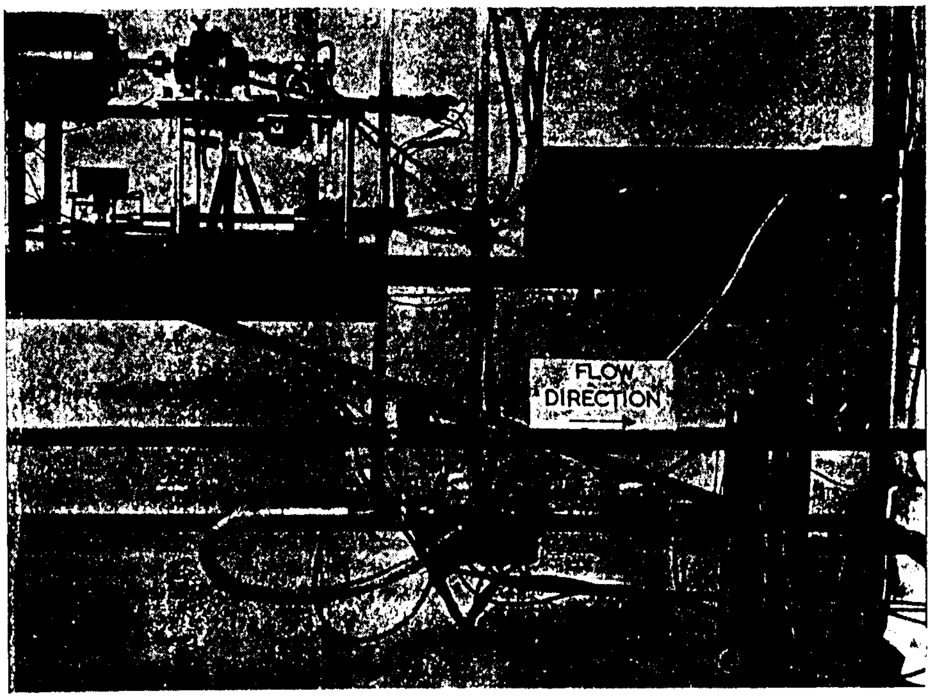

Fig. 2.—*View of the experimental pump and drive system*

resistance losses were thereby increased they could be calculated. A more efficient design, though harder to analyse, is to continue the conical shape of the jets to include the gas-liquid pistons. Such a design has been made and successfully tested using a pneumatic drive system with servo-controlled valves.[3]

Theory

It has been shown[2] by using the generalised form of the Euler momentum theorem that the relationship between pump-head and jet velocity may be expressed for a cylindrical main pipe as:§

$$h_p = nk\overline{v_{JB}^2}\alpha(1-\alpha/2-2\,v_p/\bar{v}_{JB})/g \qquad . \qquad (1)$$

In the bulb-Venturi pump the main pipe is not cylindrical but is similar to the shape shown in Fig. 1. This geometry will have little effect on the theory of the blow stroke provided that α is defined as the area ratio of jet nozzle to Venturi. However, the loss of pump-head due to the suck stroke is significantly reduced in this design because the liquid velocity in the bulbous part of the main pipe is lowered so that less velocity head is extracted. It might be argued that this should give no advantage because the pressure-head would be correspondingly increased and extracted with a similar loss of energy from the pump. However, this same pressure-head must be returned on the blow stroke so that no net change of pressure energy occurs during a complete cycle.

A shortened analysis of the theory of operation of the pump now follows. The blow and suck stroke of a single jet are analysed separately and then combined using a steady-state model of a jet of area, A_J, directing liquid of constant density, ρ, into a cylindrical mixing region of a Venturi of area, A_P. It is assumed that the body force (*e.g.* gravity) is zero and that the jet Reynolds number is greater than 2000.

The instantaneous force, F_{JB}, developed by the pump during a steady blow stroke is given by the pressure force plus the rate of increase of momentum from the input. Using the Euler momentum theorem and the continuity equation we may derive:[2]

$$F_{JB} = \rho A_J v_{JB}^2(1-\alpha/2-v_p/v_{JB}) \qquad . \qquad . \qquad (2)$$

This force acts over the area of the Venturi, A_P, so the instantaneous head developed during the blow stroke is given by:

$$h_{JB} = (\alpha v_{JB}^2/g)(1-\alpha/2-v_P/v_{JB}) \qquad . \qquad . \qquad (3)$$

The instantaneous force produced by the suck stroke is given by the rate of extraction of momentum from the main part of the bulb carrying liquid of velocity, v_B. For simplicity we will consider the bulb as being cylindrical up to the Venturi and of area A_B. This force is given by the loss of velocity-head, and when combined with the continuity equation may be written:

$$F_{JS} = -\rho A_J v_{JB} v_{JS} \qquad . \qquad . \qquad . \qquad (4)$$

This force acts over the area of the bulb, so using $\beta = A_J/A_B$ we may write the head loss during the suck stroke as:

$$h_{JS} = -\alpha\beta^2 v_{JS}^2/g \qquad . \qquad . \qquad . \qquad (5)$$

For a symmetrical variation of jet velocity during a complete cycle of blow and suck we have $\overline{v_{JB}} = \overline{v_{JS}}$ and $\overline{v^2{}_{JB}} = \overline{v^2{}_{JS}}$ for the mean and mean squared velocities respectively. The average head produced by a pump with n jets will be given by:

$$h_p = n(\bar{h}_{JB}+\bar{h}_{JS}) \qquad . \qquad . \qquad . \qquad (6)$$

Hence combining equations (3), (5), and (6) the average pump-head is given by:

$$h_p = (n\,\overline{v_{JB}^2}\alpha/g)[1-\alpha/2-(1+\beta^2)(v_P/\overline{v_{JB}})] \qquad . \qquad (7)$$

In the experiments to be described $\beta = 1/30$ and so the suck stroke has only a small effect. In a pump with a cylindrical main pipe and no Venturi $\beta = 1$.

A simple mathematical model may be used to obtain an approximate relationship between pump-head and drive pressure. This involves considering the behaviour of two piston chambers (attached to nozzle/diffusers) which feed liquid alternately from one to the other *via* the pump body. It is assumed that gravitational and inertial effects within the piston chambers are small, and that the piston area is much larger than that of the jet. By taking into account typical nozzle and diffuser losses, the instantaneous space-average of the jet blow velocity squared, available from a twin-jet combination at a given stage, may be expressed as:

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§ *Symbols have the meanings given them on p. T128*

$$v_{JB}^2(t) = 2C_J \Delta p(t)/\rho . \qquad (8)$$

The mean value including both jets may be expressed:

$$\overline{v_{JB}^2} = 2C_J \overline{\Delta p}\, \rho = (2C_J/\rho T) \int_0^T \Delta p(t)\, \mathrm{d}t \quad . \qquad (9)$$

where the integral may be determined from the average over a cycle ($T = 1/f$) of the enclosed area of a recorded trace of drive pressure. Substituting equation (9) in equation (7) with $n = 2$ and $v_p = 0$ we have the pump stagnation-head given by:

$$h_{p0} = 2C_J f\alpha \left[(1-\alpha/2)/\rho g\right] \int_0^T \Delta p(t)\, \mathrm{d}t \quad . \qquad (10)$$

Equation (10) relates h_{p0} to the measured drive pressure which may be provided by valved compressed air or, as in the present case, by reciprocating mechanical pistons. However, equation (10) does not consider how $\Delta p(t)$ is achieved, that is, it does not relate head to the sinusoidally swept volume of the mechanical pistons. This requires a more extensive treatment because the pressure is related to the results it achieves. This will now be considered below.

The relationship between jet velocity and input parameters may be derived with the help of the simplified model shown in Fig. 3. This represents a twin-jet stage comprising two extended piston chambers and jets, which transfer liquid from one to the other in anti-phase. This would occur in practice only if the inlet and outlet to the main pipe of the pump were sealed off. In fact the closest practical approach to this situation is when the pump is at stagnation [that is, when $v_p = 0$ in equation (7)]. The two cylindrical tubes of constant radius, r, which represent the two piston chambers, each have a short constriction at the end representing the nozzle/diffusers. These are directed towards each other, along a common horizontal axis, into a sealed chamber representing the pump body, in such a way that one jet does not point

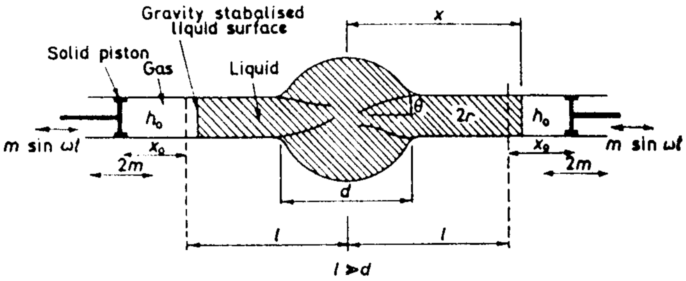


Fig. 3.—*Simplified theoretical model of a single twin-jet stage*

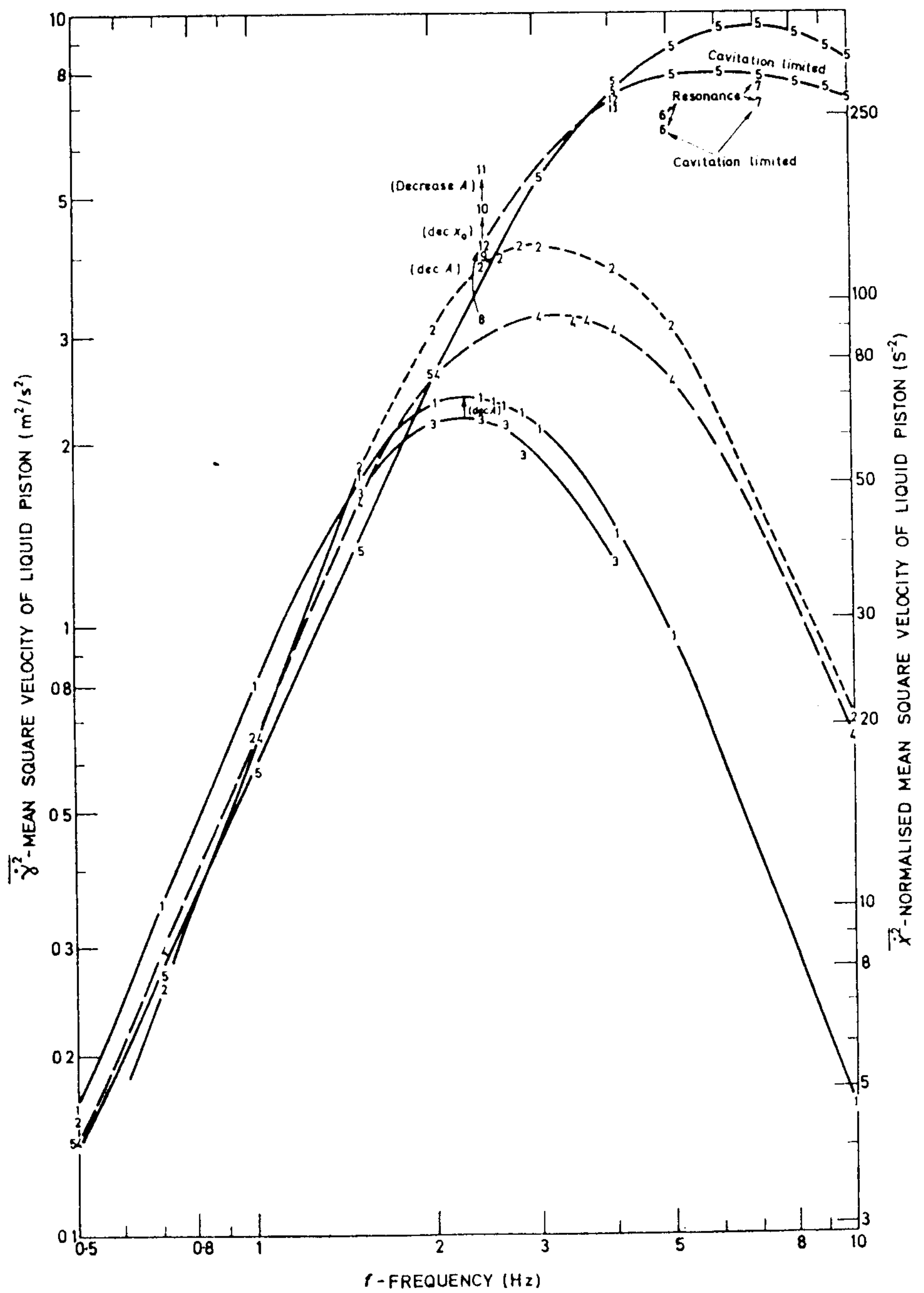


Fig. 4.—*Analogue computer solutions for the variation of liquid-piston velocity with frequency*

directly into the other. A solid piston is reciprocated sinusoidally ($m \sin \omega t$) in each of the two extreme open ends, so that their separation is constant. The liquid, of length $2l$, is located between these pistons with a gas cushion at each end, of length x_0 when at its centre position. The pressure is then p_0, which is the ambient pressure within the pump body at the chosen pump stage. The variation in pressure difference due to piston movement will cause the liquid to cycle at a frequency $\omega/2\pi$, and displacement $x(t)$. In the experimental configuration with vertical air-liquid piston chambers, the liquid surfaces are stabilised by gravity and surface tension; also the static-head differences are small compared with the drive pressures, so gravitational restoring forces can be ignored.

The equation of motion may be calculated by equating the pressure forces, which are dependent on the displacement of the liquid surface, $x(t)$, to the opposing forces due to friction, inertia, and the nozzle pressures. The equation may be reduced to the following form:

$$\ddot{X}+A\,|\dot{X}|\,\dot{X}-B\left[\frac{L+\sin\omega t-X}{X_0^2-(L+\sin\omega t-X)^2}\right]=0 \quad (11)$$

The damping factor, A, is given by:

$$A=[f/4R]+[F(\gamma-1)/2L\tan\theta]+$$
$$[(\gamma^2-1)/4L\,C_{NB}]+[1/4LC_{NS}] \quad (12)$$

The terms in equation (12) represent respectively friction at the walls, friction in the nozzle sections, the reaction due to the creation of useful pump head, and the nozzle/diffuser loss on the suck stroke. For the typical set of experimental figures, which are shown in the list of symbols, the useful-head term accounts for 83% of the damping term, which was calculated to be $A = 1{\cdot}68$. This 83% may be termed the theoretical efficiency of the drive system in converting pneumatic energy to jet velocity-head.

Equation (11) is a non-linear second-order differential equation, since the restoring force is dependent on displacement, and the damping term is proportional to $\dot{X}^2$ and changes sign with $\dot{X}$. These factors make it impossible to solve the equation in any reasonably simple analytical manner. An analogue computer was therefore used. This gave the advantage that the practical limitation of $\dot{X}$, due to cavitation at the nozzle/diffuser during the suck stroke, could be included in the theoretical analysis by adding the constraint:

$$\dot{X}^2 \leqslant 2C_{NS}\,p_0/\rho m^2(\gamma^2-1) \quad . \quad . \quad (13)$$

The pump stagnation-head is given by $v_p = 0$ in equation (7); and using $\overline{v^2_{JB}} = \gamma^2\overline{\dot{x}^2} = \gamma^2 m^2\,\overline{\dot{X}^2}$ we have:

$$h_{p0}=(nk\alpha/g)[1-\alpha/2]\gamma^2 m^2\,\overline{\dot{X}^2}. \quad . \quad (14)$$

Values of $\overline{\dot{x}^2}$ and $\overline{\dot{X}^2}$ were obtained from the computer and are shown in Fig. 4. The $X(t)$ and $\dot{X}(t)$ output traces were frequently non-sinusoidal with a large even-harmonic component. This became especially evident for large $\dot{X}$ when the cavitation constraint was switched into the computer circuit and caused up to 17% loss of maximum head.

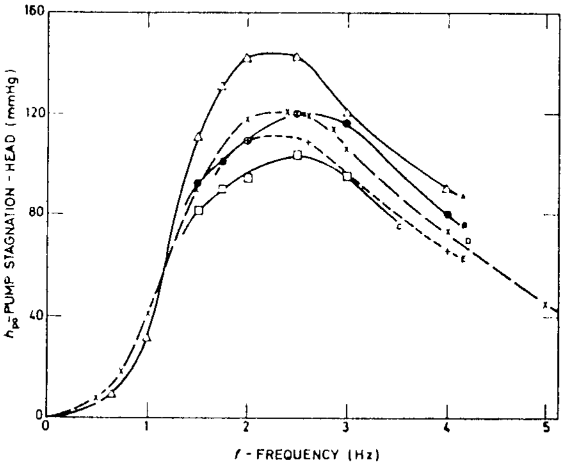
The two lower curves in Fig. 4 were used in conjunction with equation (14) to calculate the curves of theoretical head as a function of frequency, and are shown in comparison with experimental results in Fig. 5. These results are discussed later.

The variation of parameters in equation (11) can cover a wide range of possible systems by judicious selection of values. A change in diameter which would be required for the theoretical study of larger systems merely involves an increase in R in equation (12); this makes the corresponding term in the damping factor A even less important than it was before.

Description of Apparatus and Instrumentation

A plan view of the two-stage twin-jet pump used for experimental studies is similar to Fig. 1. However, in order to facilitate the measurement of jet velocity, the gas-liquid piston chambers were made in the form of long vertical cylinders and were curved through 90° as shown in Fig. 2. The pump was tested in a closed loop and circulated water through an open tank with a ball-valve located on the outlet side of the pump to control flow-rates. The mechanical drive system embodied four pistons which were reciprocated by cams driven from a ¼ h.p. motor *via* a Variator and gearbox. The Variator enabled the drive frequency to be varied between 0·6 and 5·2 Hz, while a variable fulcrum on the rocker arms allowed mechanical piston strokes of between 42 and 70 mm to be obtained. The phase difference between the two stages was set at $\pi/2$, and the difference between neighbouring jets at π.

The jet velocity was determined by measuring the alternating velocity in each of the piston chambers using a double Pitot-tube arrangement mounted through the tube walls. The differential pressure between the Pitots was recorded *via* a pressure transducer. During each half cycle one Pitot tube faced the direction of motion of the liquid and measured stagnation pressure while the other faced away and effectively measured the static pressure. The four Pitot-tube assemblies were calibrated separately using both alternating velocity at different frequencies, and constant velocity in both directions. The alternating velocities were calculated in each case from a measurement of the liquid stroke. Pressure transducers rated at 50lbf/in^2 were used to measure the drive pressures in each of the four cylinders of the mechanical drive system and all transducer signals were displayed simultaneously on a 12-channel ultra-violet recorder. Provision was made for a mercury manometer to be connected across one or both pump stages or alternatively, instantaneous pressures could be monitored by attaching pressure transducers at the same points. A flow-meter was independently calibrated and incorporated in the main pump circuit. In order to compare the phases of the liquid stroke and of the drive pressure with that of the mechanical drive system, the latter was monitored in two ways. A potentiometer coupled to the u.v. recorder was driven off one mechanical piston so as to give the instantaneous position; and while at maximum displacement this piston triggered a pulse circuit, which deflected another u.v. spot to give a timing mark.

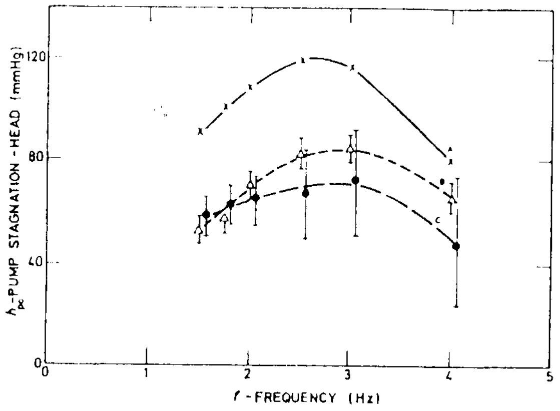


a: V_0 = 106 ml
b: V_0 = 150 ml
c: V_0 = 178 ml
d: Computer results: V_0 = 150 ml
X_0 = 5·36
A = 1·53
e: Computer results: V_0 = 150 ml
X_0 = 5·36
A = 1·7

Fig. 5.—*Measured variation of stagnation-head with frequency for different dead volumes, compared with Equation (11)*

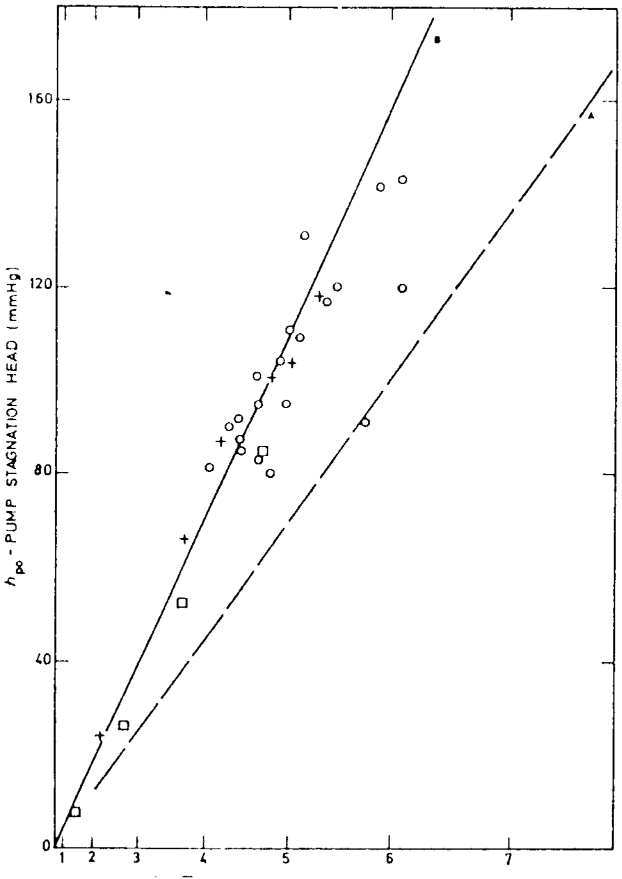
Experimental Studies

The stagnation-head, h_{p0}, of a single twin-jet stage was measured by allowing the pump to develop its full pressure-head while operating with the ball-valve closed. Figure 5 shows how the pump stagnation-head varied with frequency for three different "dead" volumes, V_0, of air enclosed between the mechanical and liquid pistons. Each curve is of a similar shape and shows a resonance peak somewhere between 2·3 and 2·6 Hz. Results computed from equation (11) for $V_0 = 150$ ml are shown for comparison. In Fig. 6 curve A shows the measured head for $V_0 = 150$ ml in comparison with two curves calculated from equation (7) with $v_p = 0$, using measurements taken at the same time. Curve B shows the stagnation-heads derived from velocities measured by Pitot tubes, and curve C shows the stagnation-heads derived from the inexact visual measurements of liquid strokes. The measured heads are seen to be up to 60% greater than those calculated using a theory which takes no account of boundary layers.



a: head measured with mercury manometer
b: head calculated from Pitot-tube measurements and equation (7)
c: head calculated from stroke in air-liquid piston chamber and equation (7)
$V_0 = 150$ ml, no boundary-layer correction

Fig. 6.—*Measured variation of stagnation-head with frequency compared with equation (7)*



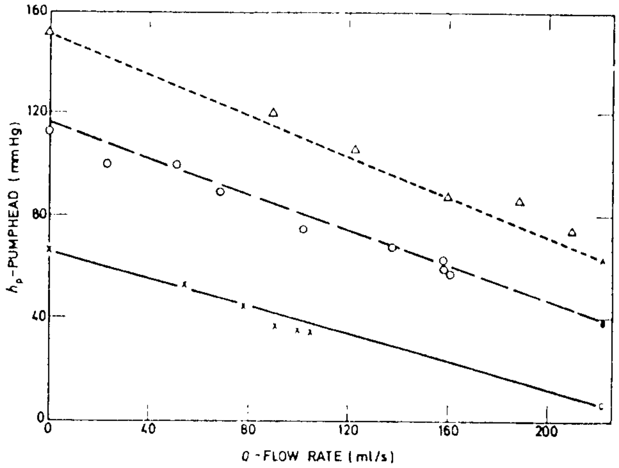
Experimental results
□: second stage of 4-jet systems
+ : sealed system
○: dead volume varied
Theory
a: theory of Equation (7): no boundary-layer correction
b: theory of Equation (7) assuming boundary layer 0·41 mm thick

Fig. 7.—*Measured variation of stagnation-head with jet velocity for a single stage compared with equation (7)*

This discrepancy is shown more clearly in Fig. 7 where the stagnation-heads are plotted as a function of v^2_{JB} as determined by Pitot-tube measurements and compared with the theoretical line A predicted by equation (7). Line B passes through the experimental results and lies nearly 60% higher than line A. Line A may be derived from equation (7) with the assumption of boundary-layer effects. This is discussed later.

Pump-head as a function of flow-rate

The variation of pump-head with flow-rate for three different jet velocities is shown in Fig. 8. The measured pressure rise across the pump was increased to compensate for the internal resistance of the pump at the same flow-rate. The internal resistance was determined in a preliminary experiment with the jets inactive by passing water through the pump and measuring the pressure drop. This had the expected square-law dependence on flow-rate and amounted to about 40 mm Hg at 0·2 l/s. Also shown are the theoretical heads calculated from equation (7) using jet velocities derived from Pitot-tube measurements with boundary-layer corrections incorporated.



v_{JB}	Theory	Experiment
a: 5·9 m/s	- - - - -	△
b: 5·2 m/s	— — —	○
c: 3·9 m/s	———	×

Theory uses measured $\overline{v_{JB}}$ substituted in equation (7), with boundary-layer correction

Fig. 8.—*Measured variation of pump-head with flow-rate for a single stage compared with equation (7)*

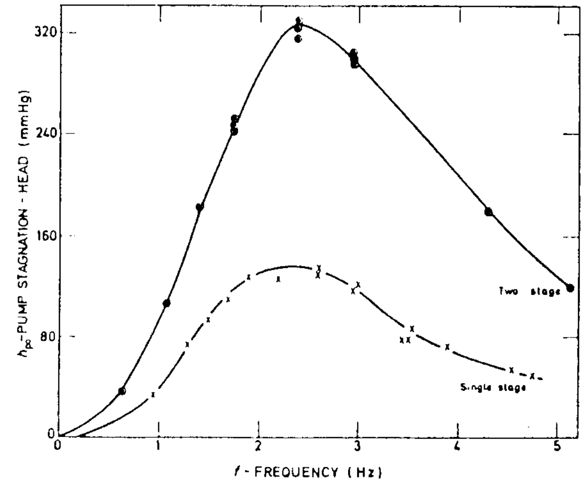
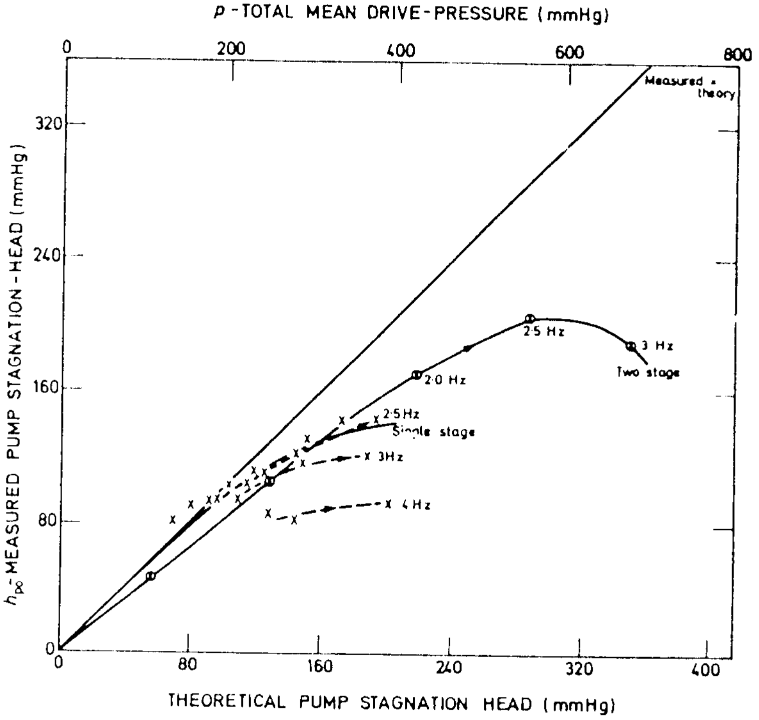


Fig. 9.—*Variation of stagnation-head with frequency for a two-stage pump.* $V_0 \sim 120$ ml



——→—— : increasing f, constant V_0

- - - → - - - - : constant f, decreasing V_0

Fig. 10.—*Comparison of measured stagnation-heads with theoretical values based on measured drive pressures and equation (10)*

Two-stage pump

Experiments similar to those already described for a single pump stage have been conducted using two stages. In Fig. 9 the variation of stagnation-head with frequency is shown, and compared with the corresponding curve for a single stage, the dead volumes in each case being similar. The maximum stagnation-head achieved by two stages was 325 mm Hg at a frequency of 2·4 Hz, which was 2·44 times the head of 133 mm Hg achieved by a single stage at the same optimum frequency and similar dead volume. This difference was attributed to the increased mean pressure produced by the first stage. This increased the jet suck velocity and hence the stroke of the second stage.

Comparison of experimental head with theoretical head based on drive pressures

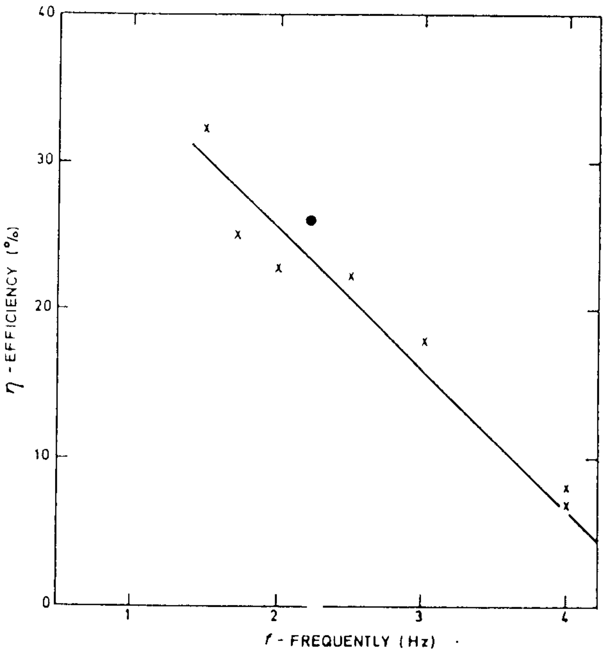
The mean drive pressure over a single cycle was derived by integration of the u.v. recorder traces for each jet. In Fig. 10 the theoretical head, based on mean drive pressure and using equation (10), is compared with the experimentally obtained head for single- and two-stage pumps. The agreement is as would be expected, with efficient conversion of drive pressure into velocity head at the lower pressures but at higher pressures there is increasing inefficiency as the drive pressures have to overcome the higher resistance losses in the air-liquid pistons. At the maximum drive pressure and frequency used the experimentally obtained head was only 50% of the head calculated from simplified theory.

Calculation of pump efficiency

The power input to each of the air-liquid pistons was measured from the area of the appropriate pV indicator loop. This was displayed on an oscilloscope and was derived from the volume-displacement potentiometer and the pressure transducer attached to the air-liquid piston chamber. A Polaroid camera mounted on the oscilloscope recorded the pV loop averaged over several cycles.

By assuming that the work done on the gas was small compared with the work done on the liquid in the piston chamber, the pumping efficiency was only slightly underestimated. The power output of the pump for a given jet velocity is proportional to the product of pump-head and flow-rate and is a maximum when $h_p = \frac{1}{2} h_{p0}$. This was calculated from the characteristics of head and flow which included internal resistance losses.

Figure 11 shows the variation with frequency of pump efficiency, based on these input and output powers. Also included in the graph is one point calculated using a more accurate pV indicator loop plotted from p and V derived from the u.v. recorder. The results are fairly scattered due to the inaccuracy in determining the input power from the pV loops; however, the overall trend is as expected, with efficiency decreasing as the frequency is raised. The maximum measured efficiency was 32% at a frequency of 1·5 Hz but in the "optimum" frequency range of 2·3 – 2·5 Hz (that is, where the greatest heads were produced) the efficiency was typically about 22%.



● : based on pV loop photographs

× : based on derived pV loop

Fig. 11.—*Variation of pump efficiency with frequency for a single stage*

The theoretical maximum efficiency of a conventional jet pump is 38%[8] with $\alpha = 0{\cdot}28$. For a reciprocating-jet pump with the same area-ratio α, it was estimated[2] that the efficiency would lie between 12 and 24%, depending on the losses during the suck stroke. However, it is found that $\alpha = 0{\cdot}5$ is quite satisfactory with the reciprocating-jet pump and moreover α close to unity is feasible. In a conventional jet pump this is out of the question, since very little of the pumped fluid could be transferred. The higher values of α permissible with the reciprocating-jet pump tend to offset losses during the suck stroke

Discussion

If the jet velocities, v_{JB}, had never been derived experimentally the agreement between measured stagnation-head and theoretical stagnation-head, based on the mechanical-piston drive parameters and pump dimensions would have been sufficiently good for no further explanation to be required. However, some values for v_{JB} were derived and differed by some 26% from those expected. The following two points must be stressed. Firstly, v_{JB} was never measured directly because of both restricted space and the necessity of avoiding interference with the jet. It was always inferred from an estimate of the total flow through the jet orifice, which was based on Pitot-tube measurements of velocity in the straight-wall section of the air-liquid piston chamber. Secondly, v_{JB} was not constant but varied across the jet diameter. It can be shown that for a given flow-rate the calculated velocity head can vary over a wide range depending on the velocity profile assumed. It is the effect of velocity profile which is likely to be the main cause of the discrepancies when v_{JB} is considered.

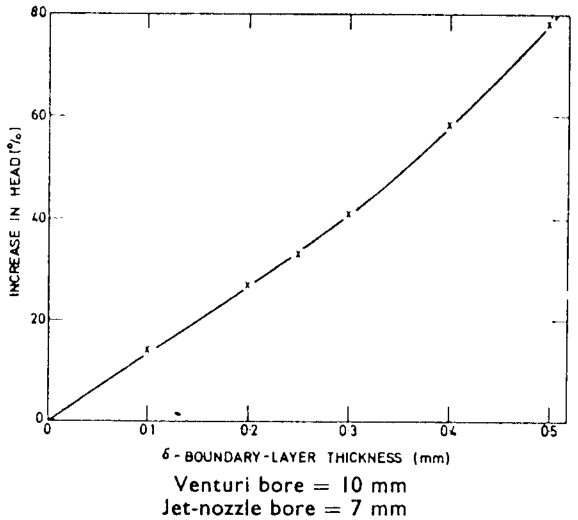


Fig. 12.—*Increase of pump-head with boundary-layer thickness for a given liquid flow-rate through the jet nozzle*

A liquid boundary-layer thickness, δ, may be defined as the thickness of zero-velocity fluid which must be present in order to provide the correct total flow and velocity head with a square velocity-profile in the central region. Figure 12 shows the effect on pump-heads calculated from equation (7) of various thicknesses of boundary layer in the orifice of the jet nozzle and Venturi. This boundary layer reduces the effective cross-sectional area of the orifices. Hence for a given liquid flow-rate through the jet nozzle, the average jet velocity is increased with a corresponding considerable increase in stagnation head. Figure 7 shows that an increase of 60% in theoretical head is required to obtain agreement with experiment. A boundary-layer thickness of 0·41 mm would provide this.

Typical jet velocities for this work were in the region of 5 m/s; at which velocity laminar boundary-layer theory gives $\delta = 0{\cdot}3$ mm, while turbulent boundary-layer theory gives $\delta = 0{\cdot}6$ mm. The theory is based on steady as opposed to oscillating flow, so the "experimental result" of $\delta = 0{\cdot}41$ mm is not inconsistent.

Figure 10 shows that experiment is in agreement with the theory of equation (10) at frequencies and drive pressures which are low but gives lower values than theory at frequencies and drive pressures which are high. The increasing discrepancy as frequency rises is because equation (10) takes no account of inertial and frictional effects in the air-liquid piston chambers. A closer agreement with equation (10) has been found when using a pump with conical piston chambers having a larger ratio of piston-to-jet area. The fact that in Fig. 10 experimental values are not greater than values predicted by a theory which takes no account of boundary-layer effects is because theoretical heads are based on measured pressure rather than jet velocity. The effect of boundary layers is merely to redistribute within the jet the total of injected momentum. This total is determined primarily by the pressure pulse, $\Delta p(t)$.

Considering now the solutions to equation (11), we see in Fig. 5 that for $V_0 = 150$ ml there is good agreement with experimental results when computed values for $\overline{X^2}$ are substituted in equation (14). The computed frequency of velocity-resonance was 2·4 Hz as compared with a measured value of 2·5 ± 0·1 Hz. However, when the pump was sealed at both ends in order to approximate more closely to the theoretical model the resonance frequency was 2·0 Hz. For the chosen system $A = 1{\cdot}68$ was calculated in equation (12) but it is seen that $A = 1{\cdot}53$ gives a value for maximum stagnation-head which equals the experimental value so the losses must be slightly less than estimated. Observed strokes were about 30% less than those predicated from equation (11). This is again due to velocity-profile and boundary-layer effects. In practice, the effect of the boundary layer is to increase γ and hence the damping term A, thus reducing the stroke. However, so far as pump-head is concerned, this is offset by increased velocity in the centre of the jet so that a similar velocity head is generated. When the two parts of the theory, namely equations (7) and (11), are put together in equation (14) the boundary-layer effect is cancelled.

A more rigorous treatment would consider the variation of boundary-layer thickness, δ, with drive pressure and velocity. In larger systems δ becomes relatively less important, so when piston-driven pneumatically-coupled reciprocating-jet pumps of larger size are being considered, the use of equations (7) and (11) is expected to give accurate predictions of pump performance.

The Strouhal numbers (defined by $St = fD/\bar{v}_J$, where f is the frequency, D is the diameter of the jet orifice, and $\bar{v}_J$ is the mean jet velocity) were typically less than 10^{-3} in this work. As the Strouhal number is increased the non-reciprocal effects will tend to disappear because vortices will have less time to develop and potential flow will take up a larger fraction of the blow stroke. In addition, the Reynolds number (defined by $Re = \bar{v}_J D\rho/\mu$, where μ is viscosity) must be high to achieve turbulent mixing during most of the blow stroke, and in this work was typically $> 10^4$. This suggests that the dimensionless number ($Re/St = \overline{v_J^{\,2}}\rho/f\mu$) should be as large as possible, say $> 10^5$. However, the limits for useful operation of the reciprocating-jet pump remain to be determined.

Conclusions

The measured characteristics of a water model of the reciprocating-jet pump agree well with theory provided the velocity profile in the jet is taken into account. Analogue computer solutions of the equations of motion of the "gas-liquid pistons" were used to derive jet velocities. These velocities were combined with momentum-mixing theory to give pump-head as a function of both flow-rate and the parameters of the mechanical drive system which provided the pneumatic energy. Measured efficiencies of up to 32% were obtained for the conversion of pneumatic energy to useful pump energy.

Acknowledgment

This work formed part of the British MHD Collaborative

Research undertaken by the Central Electricity Generating Board, the National Coal Board, the Water Tube Boilermakers' Association, and the Electrical Plant Manufacturers.

Symbols Used

A = damping factor.
A_B = area of (cylindrical) bulb.
A_J = area of jet nozzle.
A_P = area of cyclindrical mixing region of Venturi.
B = $p_0 X_0/\rho L m^2$.
D = diameter of jet orifice.
C_J = loss coefficient of nozzle/diffuser (typically = 0·67).
C_{NB} = nozzle loss coefficient when blowing (= 0·95).
C_{NS} = nozzle loss coefficient when sucking (0·7).
F = d'Arcy friction coefficient (= 0·03).
F_{JB} = instantaneous force developed by blow stroke.
F_{JS} = instantaneous force developed by suck stroke.
f = frequency of cycle.
g = acceleration due to gravity.
h_{JB} = average head produced by blow stroke.
h_{JS} = average head produced by suck stroke.
h_p = average head produced by n jets.
h_{p0} = pump stagnation-head.
k = ratio of blow period to total time of cycle, $T(= 1/f)$.
L = l/m = normalised mean length of liquid in a piston chamber (= 3·23).
m = effective amplitude of the solid piston displacement that is, as corrected by the area ratio between the mechanical and liquid pistons (0·17 m).
n = number of jets.
p_0 = equilibrium air pressure in the gas column (typically atmospheric pressure = 10^5 N/m^2).
R = r/m = normalised piston-chamber radius (= 0·03).
Re = Reynolds number (= $\overline{v_J}\, D\rho/\mu$).
St = Strouhal number (= $_f D/\overline{v_J}$).
v_B = liquid velocity in main part of bulb.
$\bar{v}_J$ = mean jet velocity.
$\overline{v_{JB}}$ = mean jet velocity during blow period.
$\overline{v^2_{JB}}$ = mean square jet velocity during blow period.
v_{JS} = velocity during suck period.
v_p = liquid velocity in the Venturi of main pipe.
X = x/m = normalised displacement.
X_0 = x_0/m = normalised and area corrected length of gas column in the equilibrium position when $\sin \omega t = 0$ (that is, $x_0 = 0{\cdot}91\ m$ corresponds to a dead volume V_0 = 150 ml in the experiments).
x = displacement of liquid.
$x(t)$ = displacement of liquid surface.
α = ratio of area of jet nozzle to area of Venturi or main pipe.
β = A_J/A_B.
γ = ratio of air-liquid piston area to jet area (= 4·28).
$\Delta p(t)$ = modulus of instantaneous difference in gas pressure in the two piston chambers.
μ = viscosity.
θ = half angle of the conical nozzle (= 10°).
ρ = density of liquid.
ω = frequency.

Except when numerical values are inserted, the above quantities may be expressed in any set of consistent units in which force and mass are not defined independently.

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March 26, 1968 J. R. STUTELY ETAL **3,374,743**

PUMPS

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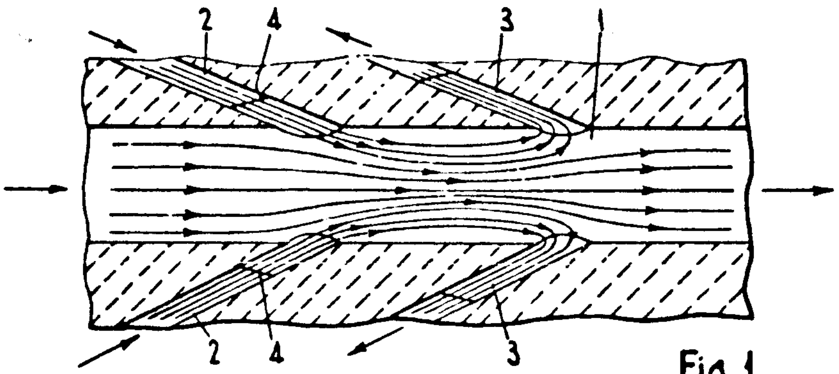

Fig. 1

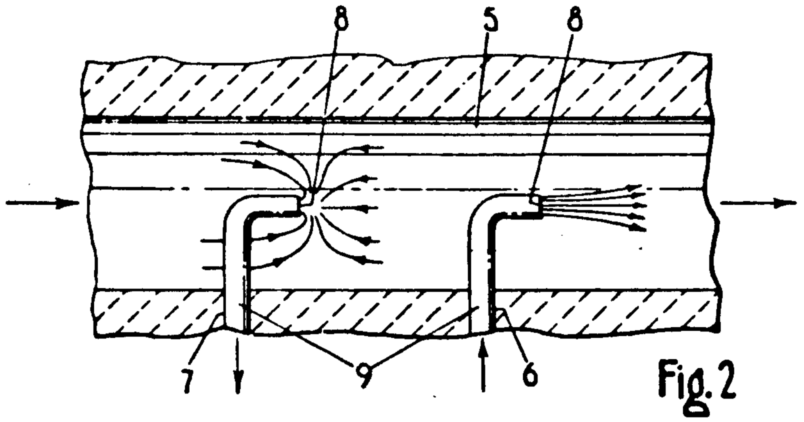

Fig. 2

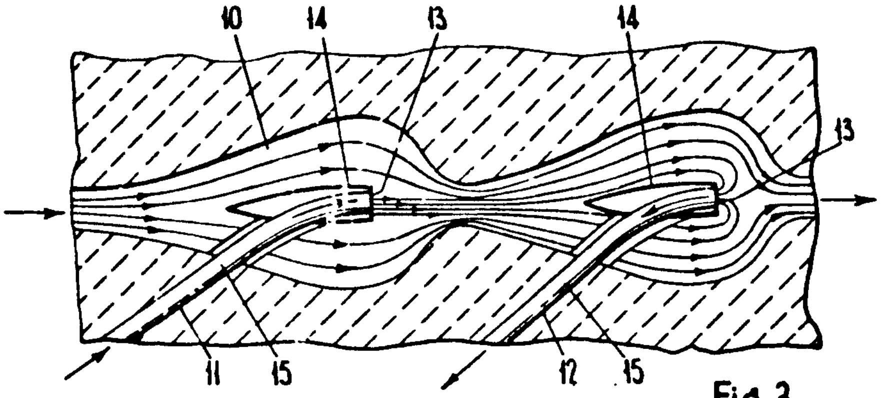

Fig. 3

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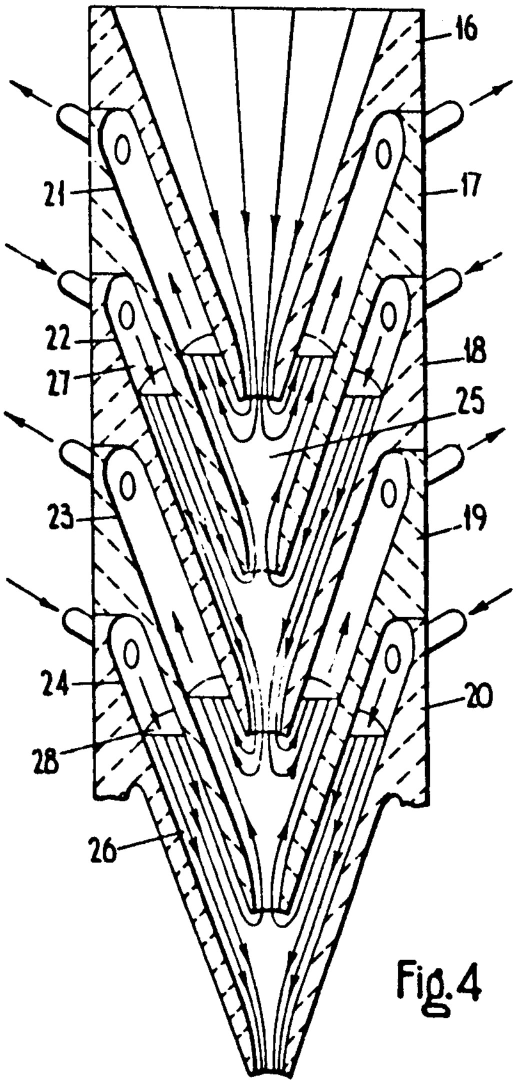

Fig.4

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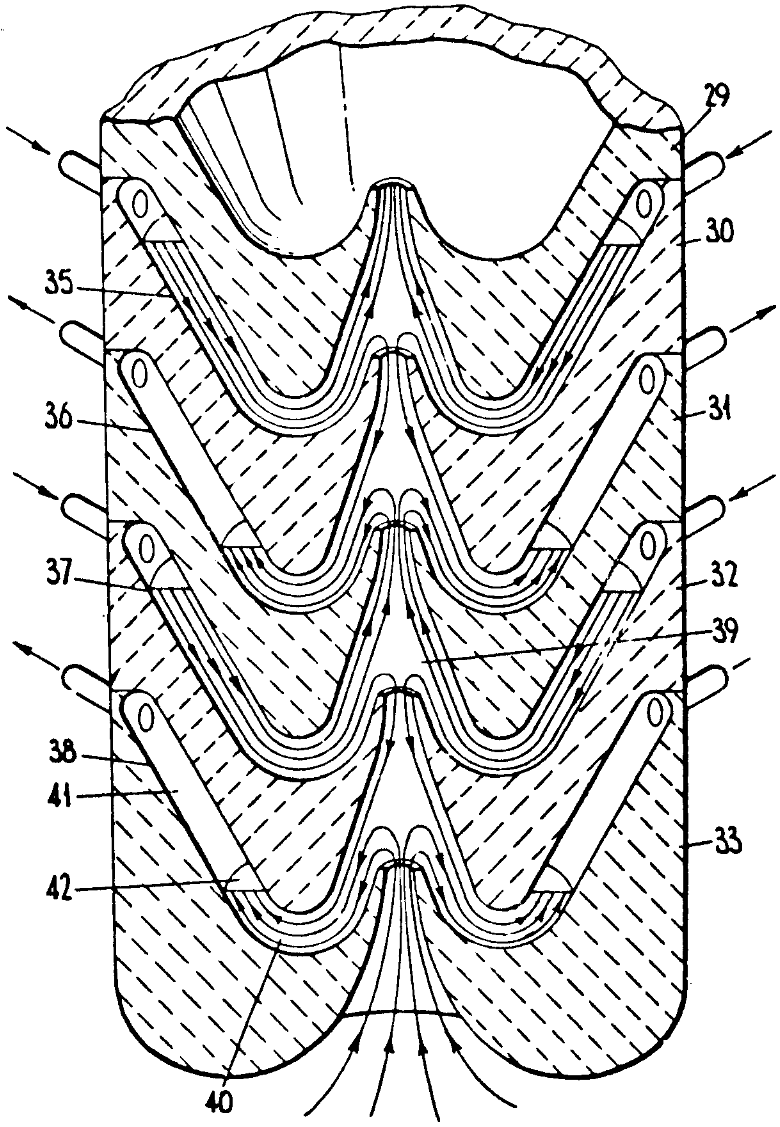

Fig. 5

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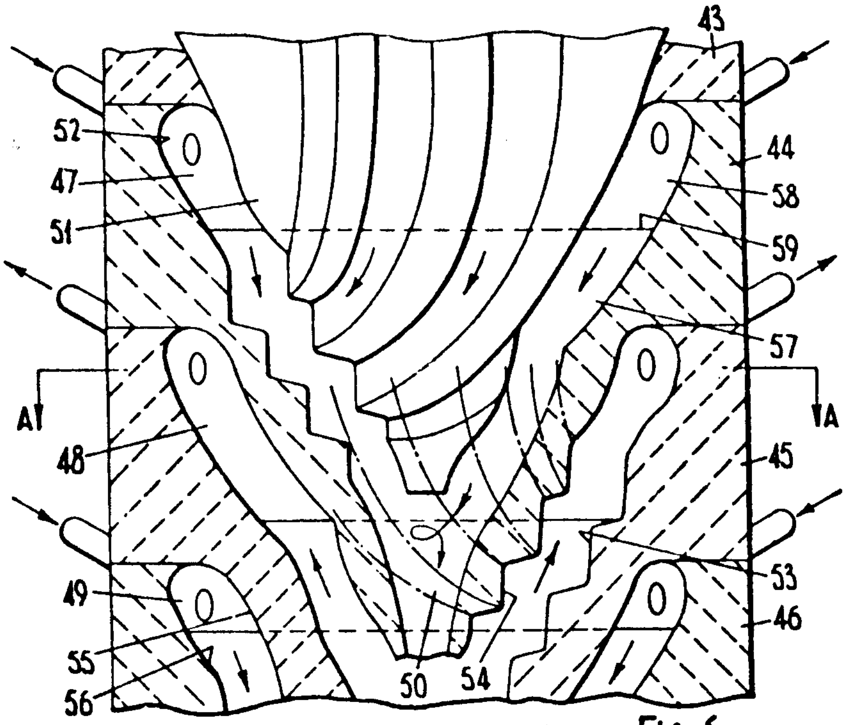

Fig. 6

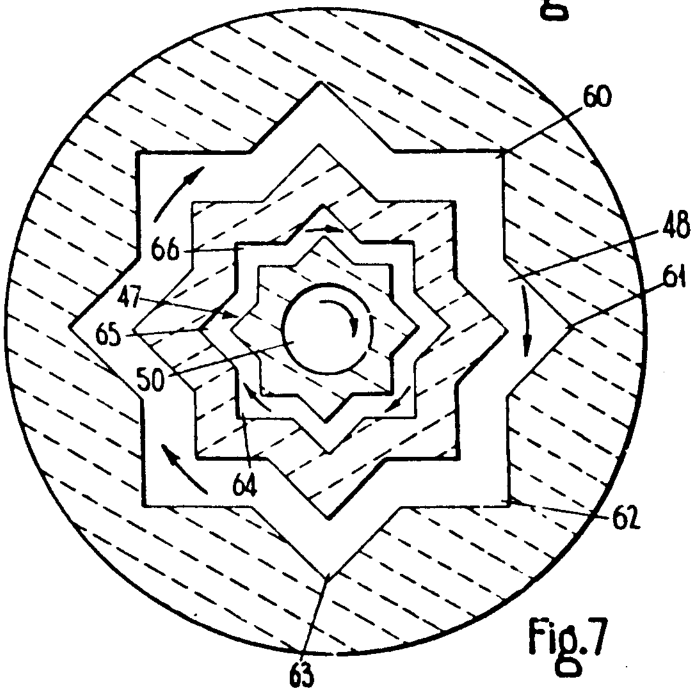

Fig. 7

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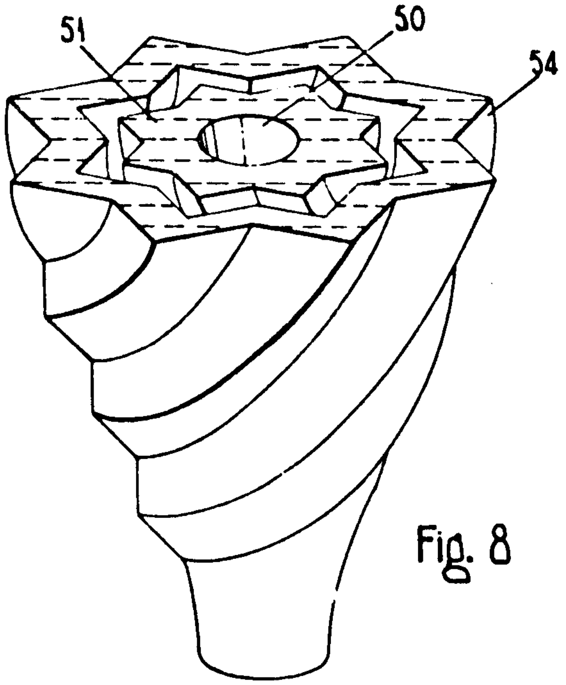

Fig. 8

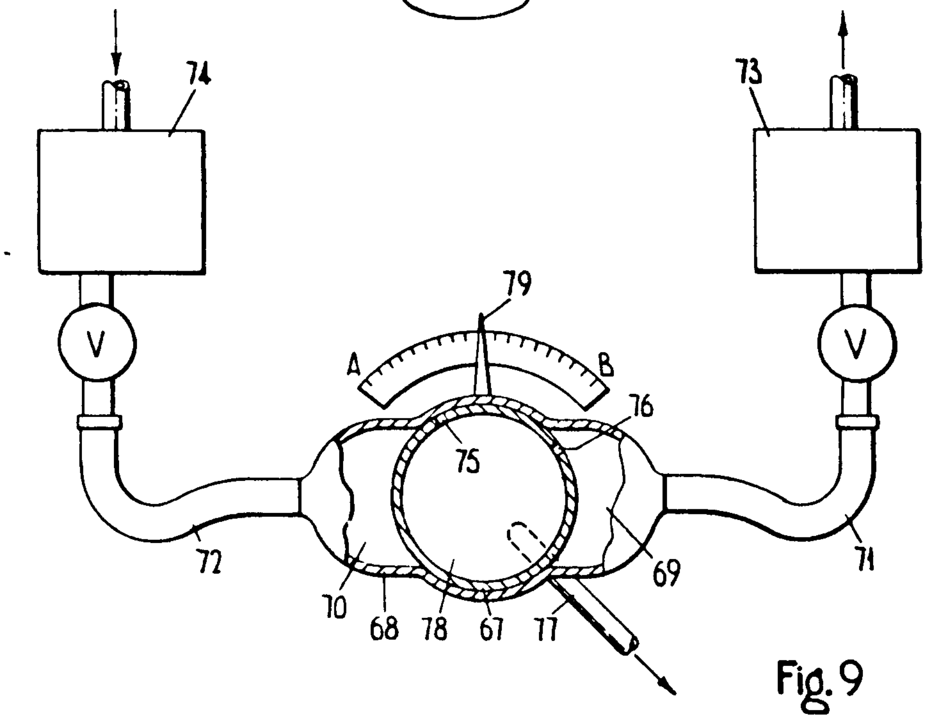

Fig. 9

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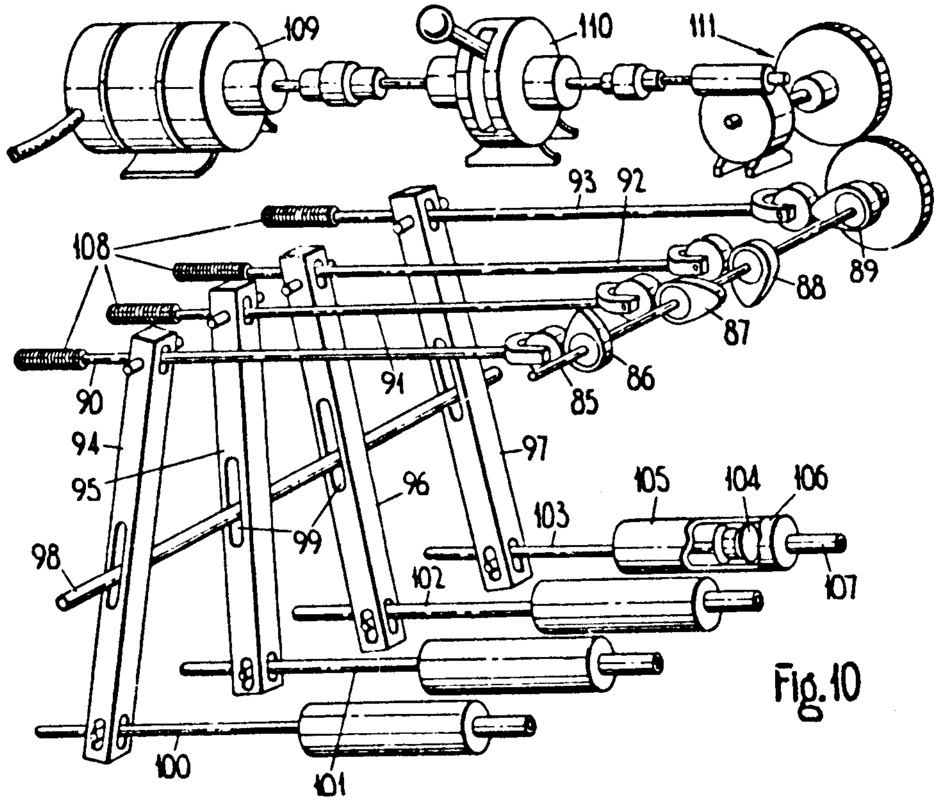

Fig. 10

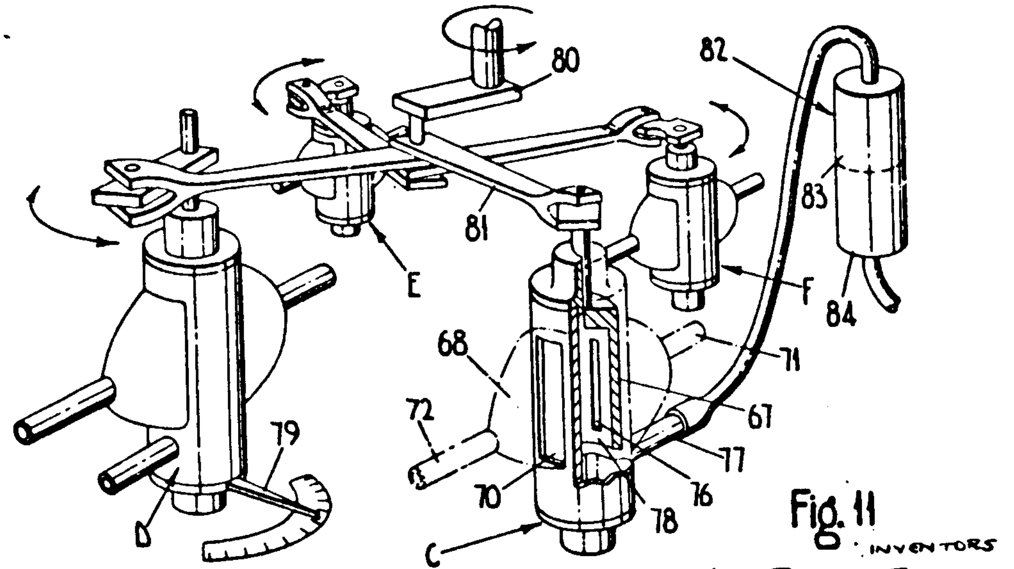

Fig. 11

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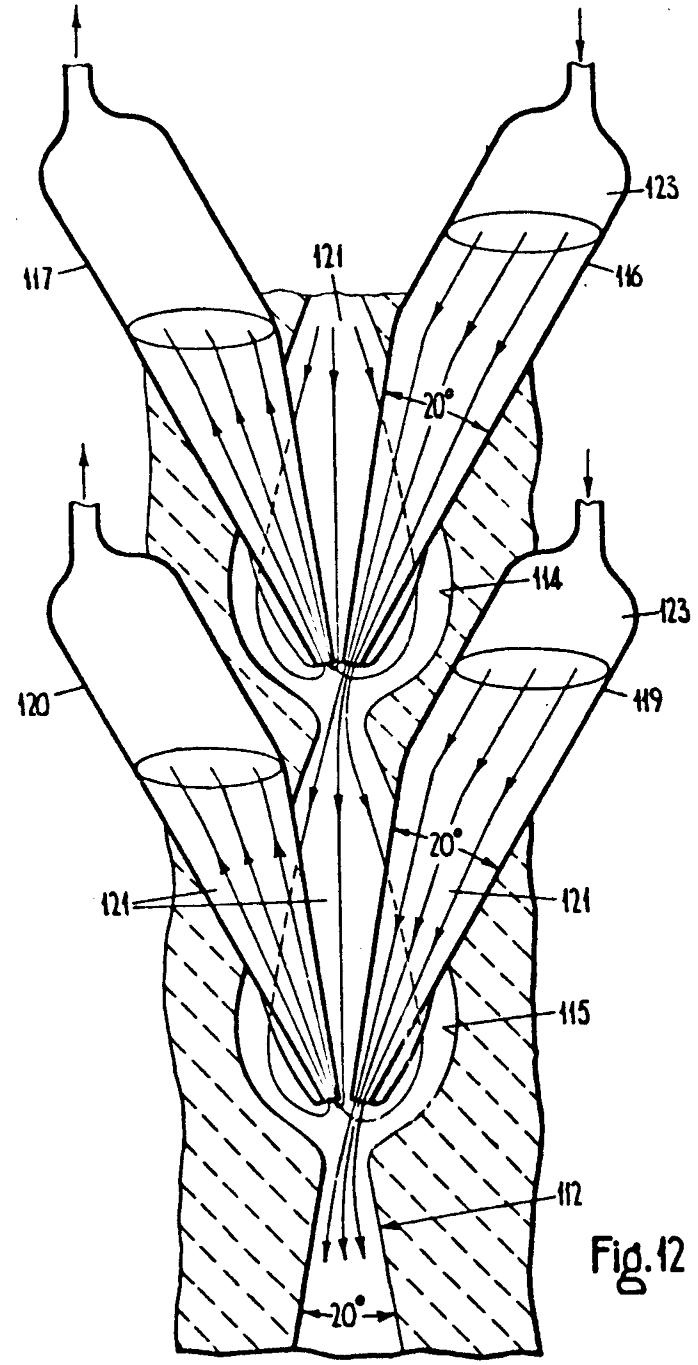

Fig. 12

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United States Patent Office

3,374,743

Patented Mar. 26, 1968

3,374,743

PUMPS

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Filed Aug. 24, 1966, Ser. No. 574,758
Claims priority, application Great Britain, Sept. 2, 1965, 37,530/65
16 Claims. (Cl. 103—1)

ABSTRACT OF THE DISCLOSURE

A pump especially suitable for use with corrosive fluids has a fluid-filled pump channel with side passages extending from points spaced along the channel length, each of which passages is partly filled with the fluid the amount of which lying within the passage is caused to pulsate by means entirely external of the channel, the passage inlets being directed along the channel length and the pulsations between the passages being so phased as to induce a net forward flow of the fluid. A liquid can be pumped by variations in the gas pressure applied over the liquid level within the passages, and the passages and passage inlets can be arranged so that for each passage the suck strokes as well as the blow strokes induce a forward flow of the liquid.

This invention relates to a form of pump for the pumping of fluids and more particularly, but not exclusively, for the pumping of high temperature and/or corrosive fluids.

Conventional pumps such as those relying on the direct pumping action of pistons or turbine blades in contact with the fluid being pumped have the disadvantage in the handling of high temperature and/or corrosive fluids of severe corrosion together with the possible melting of parts and may even prove impossible to use for the pumping of some fluids of this nature.

It is an object of the present invention to provide a pump which can be designed to be more suitable for use with at least some fluids of this nature.

According to the invention, a pump for fluids comprises a channel from which there project a plurality of passages spaced in succession along the length of the channel, the channel extending from a first region to a second region and the passages being connected to means for effecting pulsations of fluid along the passages when, in use of the pump, the channel is filled with fluid which extends into the passages, said means being arranged to so phase the pulsations between the various passages, and the passage inlets being so directed along the length of the channel, as to induce a net forward flow of fluid along the channel from said first to said second region when these regions are respectively at low and high pressures.

A pump in accordance with the invention can be so constructed that the pulsations of fluid along the passages are produced by variations in the pressure of a gas or liquid linkage such that there are no mechanical moving parts such as pistons or turbine blades which might be liable to corrosion or melting directly in contact with the fluid being pumped.

A pump in accordance with the invention can be so designed that there is a net forward flow of fluid induced along the channel during both the suck and the blow strokes of the pulsations in any passage.

In one form of the invention the channel is of approximately cylindrical shape and the passages are in the form of tubes extending from the walls of the channel and inclined backwards with respect to the direction of flow of the fluid.

In another form of the invention the channel is of the

form of a series of venturi parts and the passages extend as tubes through the walls to the centre of the channel and within the channel are bent in such a way that their nozzles point along the axis of the channel in the desired direction of pumping.

In a particular type of this last form for pumping liquids downwards across high pressure gradients the said channel is vertical and two passages project into each venturi, one on opposite sides.

In yet another form of the invention the channel is vertical and each passage of said channel is approximately funnel-shaped, the aperture of each passage where it joins the channel extending round the circumference of the channel.

In a pump of the last mentioned kind arranged for pumping liquids downwards the passage funnels point downwards, the pressure responsive medium is a gas and the surfaces of each funnel passage are preferably ridged internally in spirals about the axis of the central channel in such manner as to cause the liquid in each passage on a blow stroke to swirl into a vortex motion within the passage and thereby upon ejection, by virtue of the centrifugal force acquired, to be flung outwards and down the channel rather than upwards.

Additional features which may be incorporated in said last mentioned kind of pump in order to enable the pumping of liquids upwards to be effected is the provision of funnel passages each of which is upwardly pointing adjacent the channel but bent back upwardly upon itself to provide a bend at the bottom communicating between downward and upward funneled passage portions and drive means so adapted as to effect pulsations in which the pressure responsive medium fluid interfaces oscillate wholly within the downward portions of the passages so that no pressure responsive medium is blown out into the central channel and so lost.

In a pump of said last mentioned kind and with said spirals it is preferable for the spiralling within each funnel passage to be of the opposite clockwise sense to those in the passages immediately above and below it, thereby promoting, by angular momentum coupling, more efficient transfer of fluid out of one passage on a blow stroke and into the next in the desired direction of pumping.

In a pump of said last mentioned kind for pumping liquids the axis of each passage and the axis of the central channel in the direction of pumping are preferably inclined acutely and the edge of the nozzle therebetween in these said directions is sharp so that on a suck stroke the non-reciprocity of flow of a liquid forced around a sharp bend is utilized and liquid drawn into each said passage is effectively lodged there until the coming of the next blow stroke when optimum force of blow-out will be achieved.

The materials of which the walls of the channel and the passages are constructed or with which they are lined must of course be sufficiently refractory and chemically resistant to the fluid being pumped as not to deteriorate unduly rapidly in use.

The modus operandi of any of the above kinds of pump in accordance with the invention resides in the completely different flow patterns which arise during the blow and the suck phases of a cycle or pulsation. This results in a net transfer of momentum in the direction towards which the jet orifice of each passage is pointed. During a blow stroke a jet orifice acts as a source of directed momentum, resulting in a vector "jet flow" pattern. During a suck stroke a jet orifice acts as a sink which extracts the same mass of fluid as had been injected into the central channel in its previous blow stroke. A scalar "potential flow" pattern is formed so that the extracted fluid contains little directed momentum until it re-enters the jet orifice.

Considered in terms of energy it may be seen that during a blow stroke gas pressure energy or mechanical energy is converted initially into kinetic energy of fluid within each passage. This is vectored into the central channel where momentum mixing occurs with an unavoidable energy loss to produce an increase in fluid pressure energy. (A venturi reduces this mixing loss.) A small amount of pressure energy is lost during a suck stroke.

Consideration of the Euler momentum theorem combined with an impulse treatment, gives the following expression for the mean head developed by the pump:

$$h_p=\frac{nK<\hat{\nu}_{JD}}{g}[1+\frac{\alpha}{2}-\hat{\nu}_T/\hat{\nu}_{JB}]$$

where

n is number of jets
K is the time fraction of blow during one cycle
$\alpha\gtrsim 1$ is the ratio of jet area to tube or venturi area
$\hat{\nu}_{JD}$ is the root mean square jet velocity during blow
$\hat{\nu}_{JB}$ is the mean jet velocity during blow
ν_T is the mean flow velocity of the main throughput in the mixing region
g is the acceleration due to gravity

One use for pumps in accordance with the invention which is particularly envisaged is the pumping of hot molten slag-seed mixture in a combined heat exchanger and seed recoverer of magneto hydrodynamic generator plant.

Pumps in accordance with the invention might however also find use, for example, for the pumping of molten metal or other materials, for example in the glass and metal industries, or for the pumping of highly corrosive liquids and/or gases in the chemical industry, or for the pumping of blood in surgical operations since these pumps lack such mechanical moving parts as valves opening and shutting or pistons scraping against cylinder walls which would be damaging to the blood corpuscles.

Several arrangements in accordance with the present invention will now be described, by way of example, with reference to the accompanying drawings, in which:

FIGURE 1 is a sectional side view of part of a pump according to the invention for pumping liquid at a high rate of flow,

FIGURE 2 is a sectional side view of part of a further pump according to the invention for pumping liquid,

FIGURE 3 is a sectional side view of part of another pump according to the invention designed for pumping liquid across high pressure gradients,

FIGURE 4 is a perspective section of part of another pump according to the invention designed for pumping liquid downwards,

FIGURE 5 is a perspective section of part of another pump according to the invention designed for pumping liquid upwards,

FIGURE 6 is a part-sectional view of part of another pump according to the invention for pumping liquid,

FIGURE 7 is a sectional view across AA of the pump shown in FIGURE 6,

FIGURE 8 is a perspective view of a fluted conical surface within the pump shown in FIGURE 6,

FIGURE 9 is a diagrammatic representation of any one of the four valves of the pneumatic drive system shown in FIGURE 11,

FIGURE 10 is a perspective view of a mechanical drive system designed more particularly for use with any of the pumps shown in FIGURES 4, 5 and 6,

FIGURE 11 is a perspective view of a pneumatic drive system designed more particularly for use with any of the pumps in FIGURES 1, 2 and 3,

FIGURE 12 is a sectional side view of part of another pump according to the invention designed for pumping liquid downwards across high pressure gradients.

Referring now to the drawings, FIGURE 1 shows in side section part of the operative length of a pump for pumping liquids designed for a high rate of flow and having a cylindrical central channel **1** provided with tubular side passages inclined backwards with respect to the direction of flow of the liquid along the central channel. Such a pump is particularly suitable for the pumping of liquids at high Reynold's numbers. Both the channel and the side passages **2**, **3** are filled with liquid **4** extending to a vertical cylinder, not shown, but to be later described (appearing as **82** in FIGURE 11), which cylinder is filled in the lower part with liquid and in the upper part with gas, the pressure of which gas is made to oscillate by external pumping means, also later described, thus causing the gas-liquid interface in the cylinder to rise and fall and liquid to flow in and out of the passages, and thereby to drive the pump. The advantage in employing oscillatory means of this kind in this and all further embodiments to be described is that more efficient coupling of momentum from the gas into the liquid is achieved than would be possible with say a continuous stream of gas which would result in bubbling of gas through the liquid and consequently small transfer of momentum from gas to liquid. The arrows indicate the lines of flow of the liquid when the passages **2** are on their blow strokes and the passages **3** are on their suck strokes. This pump and all further embodiments to be described are in particular directed to the pumping of molten-slag-seed mixture across pressure gradients of up to 10 atmospheres at temperatures between 1400° C. and 1800° C. in a combined heat exchanger and seed recoverer of magnetohydrodynamic generator plant.

FIGURE 2 shows in side section part of the operative length of a pump having a cylindrical central channel **5**. The side passages in this embodiment comprise tubes **6** and **7** which project into the channel wall perpendicularly and pass to the centre where they are bent so that their jets **8** point along the axis of the channel in the direction of pumping. Both the channel and the tubes are filled with liquid extending to a vertical cylinder, not shown, but to be later described (and appearing as **82** in FIGURE 11) filled in the lower part with liquid and in the upper part with gas, the pressure of which gas is made to oscillate by external pumping means, also later described, thus causing the gas-liquid interface in the cylinder to rise and fall and liquid to flow in and out of the passages and thereby to drive the pump. These oscillations effect an approximately scalar influx of liquid **9** into any passage upon its suck stroke and a vector efflux out of the jet of any passage along the channel in the desired direction of pumping of the liquid upon the blow stroke of the passage. The arrows indicate the lines of flow of the liquid when the passage **6** is on its blow stroke and the passage **7** is on it suck stroke.

FIGURE 3 shows in side section part of the operative length of a pump wherein to achieve pumping across large pressure gradients the channel **10** is of the form of a series of venturi parts. Tubular passages **11**, **12** of a suitable material project into the channel through the walls and pass to the centre where their jets **13** point along the axis of the channel in the desired direction of pumping. The tubular passages are enclosed in a sheath **14** for streamlining and to protect the material of the passages from erosion. Both the channel and the side passages are filled with liquid extending to a vertical cylinder, not shown, but to be later described (appearing as **82** in FIGURE 11) filled in the lower part with liquid and in the upper part with gas, the pressure of which is made to oscillate by external pumping means, also later described, thus causing the gas-liquid interfaces in the cylinder to rise and fall and liquid to flow in and out of the passages and thereby to drive the pump. These oscillations effect an approximately scalar influx of liquid into any passage upon its suck stroke and a vector efflux out of the jet of any passage and along the channel in the desired direction of pumping of the liquid upon the blow stroke of the passage. The arrows indicate the lines of flow of the liquid when the passage **11** is on its blow stroke and the passage **12** is on its suck stroke.

FIGURE 4 shows in perspective section part of the operative length of a pump for pumping liquids downward. The pump is constructed in cone shaped portions **16**, **17**, **18**, **19**, **20** which are fitted together one within the other to form the pump. The passages **21**, **22**, **23**, **24** between the portions so fitted together are funnel-shaped and lead into the central channel **25** of the pump and their cones point in the desired direction of pumping. The passages contain partly liquid **26** and partly gas **27**. The pressure of the gas **27** within each passage is made to oscillate by external pumping means later described so as to cause the gas-liquid interfaces **28** to oscillate within each passage. The arrows indicate the lines of flow of the liquid when the conical passages **22**, **24** are on their blow strokes and the conical passages **21**, **23**, are on their suck strokes. This type of pump is particularly suitable for the pumping of liquids of high to low Reynold's numbers.

FIGURE 5 shows in perspective section part of the operative length of a pump for pumping liquids upwards. The pump is constructed in rotationally symmetrical W-section portions **29**, **30**, **31**, **32**, **33** which are fitted together one upon the other to form the pump. The passages **35**, **36**, **37**, **38** between the portions so fixed together each has the form of an upwardly pointing cone with an axially symmetrical upwardly pointing U-bend in its surface. These passages open at an upwardly pointing angle into the central channel **39** up which liquid is desired to be pumped. Each passage contains partly liquid **40** and partly gas **41**. The pressure of the gas **41** within each passage is made to oscillate by external pumping means later described so as to cause the gas-liquid interfaces **42** to oscillate within each passage. The arrows indicate the lines of flow of the liquid when the conical passages **35**, **37** are on their blow strokes and the conical passages **36**, **38** are on their suck strokes.

FIGURE 6 shows in part section and in part raised elevation part of the operative length of a pump for pumping a liquid downwards. The pump is constructed in cone shaped portions **43**, **44**, **45**, **46** fitted one upon another to form the whole pump. The passages **47**, **48**, **49** therebetween the portions so fitted together are funnel shaped and lead into the central channel **50** of the pump and their cones point in the desired direction of pumping. The interior surfaces **51**, **52**, **53**, **54**, **55**, **56** of the passages are ridged spirally and for clarity **51** is shown in raised elevation and is further shown in FIGURE 8. The spiralling of surface **54** shown chain-doted is intended to illustrate the spiralling at the front of this conical surface. The spiralling of adjacent passages is indicated to be in the opposite sense to one another as is preferred. Each passage contains partly liquid **57** and partly gas **58**. The pressure of the gas **58** within each passage is made to oscillate by external pumping means, later described, so as to cause the gas-liquid interfaces **59** to oscillate within each passage. By virtue of the spiralling a vortex motion of the liquid is created within the passage upon a blow stroke so that the liquid upon ejection is by centrifugal force flung outwards and down the channel in the desired direction of pumping rather than back upwards. The arrows indicate the lines of flow of the liquid when the conical passages **47**, **49** are on their blow strokes and the conical passage **48** is on its suck stroke.

FIGURE 7 is a sectional view across AA of the embodiment shown in FIGURE 6 and in which sectional view kinks **60**, **61**, **62**, **63**, etc. in the passage **48** and kinks **64**, **65**, **66**, etc. in the passage **47** are clearly shown. The arrows indicate the senses of swirl of the liquid in the passages **47**, **48** if the spiralling of the walls of adjacent passages in FIGURE 6 are in the opposite sense of rotation (looking in the direction of pumping) and when passage **48** is on its suck stroke and passage **47** is on its blow stroke.

FIGURE 8 shows a perspective view of the fluted inner conical surface **54** within the embodiment shown in FIGURE 6 and also to some extent the fluted conical surface **51** and the central channel **50**. It can be clearly seen from FIGURE 8 that the spiralling of conical surfaces **51** and **54** in FIGURES 6, 7, 8 is in the opposite sense as is preferred.

FIGURE 12 shows in side-section part of the operative length of a pump of suitable refractory material wherein to achieve the pumping of high temperature liquid slag-seed mixture downwards across high pressure gradients in M.H.D. generator plant the channel **112** is vertical and comprises a series of venturi-parts, such as **114** and **115**, connected end-to-end. Pairs of tubular passages such as **116**, **117** and **119**, **120** project downwardly into the venturi, one pair into each venturi, respectively **114** and **115** in these latter cases, and with each passage inclined at an angle of approximately 10° to the axis of the channel. The tubular passages are tapered substantially towards their nozzles and at an angle of 10° to their respective central axes. The end-edges of the nozzles of each pair of passages are rounded to effect a smoother flow of liquid into and out of their apertures than would otherwise be obtained and the nozzles are positioned just upstream of their respective adjacent venturi restrictions. The passages contain partly liquid **121** and partly gas **123**. In use the pressure of the gas **123** is made to oscillate in each passage tube by external pumping means to be later described. These oscillations are so phased in the various passage tubes as to be exactly in phase in the passages such as **116** and **119** on one side of the central channel and exactly 180° out of phase with those such as **117** and **120** on the opposite side of the channel. The pumping means used and to be later described is arranged in this embodiment to cause the gas-liquid interfaces in the passages to oscillate wholly within the purely cylindrical portions of these passages. The arrows indicate the lines of flow of the liquid when the tubular passages **116**, **119** are on the early part of their blow strokes and the tubular passages **117**, **120** are on the early part of their suck strokes.

In any of the embodiments shown in FIGURES 1–8 or in FIGURE 12 the material employed for the walls of the channel and the passages or lining thereof may suitably be low porosity magnesium oxide, zirconium oxide or thorium oxide or other refractory material inert to the liquid being pumped.

FIGURE 9 is a diagrammatic representation of one of the valves of the pneumatic drive means of FIGURE 11 designed for use with any of the pumps shown in FIGURES 1, 2, 3 and 12. Such a pneumatic drive means is shown in perspective in FIGURE 11. FIGURE 9 shows a single valve which controls the suck and blow strokes in a single passage of any of the pumps. In FIGURE 9 a hollow metal cylinder **67** is turnable about its axis and is situated within a metallic casing **68** having diametrically opposite outlet chambers **69**, **70** formed in its walls each outlet chamber leading by way of flexible tubing **71**, **72** respectively through controllable subsidiary valves V to low and high gas pressure reservoirs **73**, **74** respectively (the gas being, of course, a non-corrosive one). The central hollow cylinder **67** has slotted ports **75**, **76** cut in its walls and situated at 90° apart from each other. These slotted ports **75**, **76** are very much narrower than the chambers **69**, **70**. Up into the bottom of the metallic casing **68** at a level lower than the extent of the hollow metal cylinder **67** there protrudes a pipe **77** leading to one of the pump passages via the cylinder **82** in FIGURE 11.

In operation of the valve, cylinder **67** is oscillated through an angle of about 90° so that ports **75**, **76** are alternately open to their respective chambers **70**, **69** and for a small fraction of the time neither is open. During the time that port **75** is adjacent to the chamber **70**, gas will rush into the interior **78** of the metal cylinder **67** and then downwards into the lower part of the metal casing **68**, into the pipe **77** and so to the gas-liquid interface for the passage being pumped and so producing a blow stroke in this arm. Likewise during the time that port **76** is adjacent the chamber **69** gas will be drawn out of the interior **78** of the metal cylinder **67** and sucked upwards from the lower part of the casing **68**, sucked out of pipe **77** so causing a suck stroke in the passage being pumped. During the times that neither of the ports **75**, **76** is open the interior of cylinder **67** and the lower part of the casing **68** will be cut off from both of the gas pressure reservoirs **73**, **74**. This period should preferably be made as brief as possible.

The metallic casing **68** is made to some extent angularly adjustable for the purpose of controlling the relative timings of the suck and blow strokes. The pointer **79** is indicative of the angular setting of the metal casing **68** and is shown in FIGURE 9 as it would be for suck and blow strokes of equal duration. The casing **68** could be set such that pointer **79** is anywhere between A and B however. With pointer **79** close to the extreme position A there would be very short blow strokes but full suck strokes and vice versa when point **79** is closed to extreme position B. This facility may be used in a system to servo-control the liquid heights in the pump passages in for example a pump for pumping liquids downwards.

FIGURE 11 is a perspective view of the pneumatic drive means, one valve of which is shown diagrammatically in FIGURE 9. In FIGURE 11 four such valves C, D, E, F of a pneumatic drive means are shown operated by a single drive **80** which rotates continuously but yet causes each of the valve arms **81** and therefore the valve interior cylinders **67** adjoined merely to oscillate continuously through approximately 90° as required. Each valve is connected by way of pipe **77** to its own pump passage via its own intermediary cylinder **82** containing partly gas **83** and partly liquid **84**. This avoids both the risk of blowing gas into the central channel in the event of an over-blow and the danger of sucking back corrosive liquid into the drive means in the event of an over-suck or of the evolution of a blockage in the central channel. In the case of embodiments 4, 5, 6 the tube **77** will be connected direct to the passages of the pump.

FIGURE 10 is a perspective view of a mechanical drive means alternative and preferred to the pneumatic drive means last described. In FIGURE 10 a cam shaft **85** carrying cams **86**, **87**, **88**, **89** drives push rods **90**, **91**, **92**, **93** which drive levers **94**, **95**, **96**, **97** pivoted about a variable stroke pivot **98** passing through slots **99** and which levers, connected at their further ends, drive further push rods **100**, **101**, **102**, **103** and which further push rods each drives its own plunger such as **104** in its own blow-suck cylinder **105**, partially filled with non-corrosive gas **106** and leading via conduit **107** to the passage in question controlled to create the necessary pumping pulsations therein. The push rods **90**, **91** **92**, **93** after being forced away from the cams by their action are later returned to their original positions by the action of the return springs **108** so that each push rod lever, further push rod and plunger oscillates causing oscillations in the passages of the pump. The phasing ot the oscillations in the various passages is governed by the relative settings of the cams **86**, **87**, **88**, **89** and these are made such that the required phasing of the pulsations in the passages of the pump are obtained. The amplitude of the pulsations in all four pump passages may be suitably increased by raising, or decreased by lowering, the variable stroke pivot **98** in the slots **99**. The camshaft may suitably be driven by an electric motor **109** through a variable speed gearbox **110** and a reduction gearbox **111**.

The drive means of FIGURES 9 and 11 could also be used with the embodiments shown in FIGURES 4, 5, 6 by connecting the pump passages of the latter to the cylinders **82** and allowing these passages and connecting means to be filled with the liquid being pumped. Alternatively the drive means of FIGURES 9 and 11 could be modified for this purpose by omitting the cylinders **82** and connecting the pipes **77** each to the corresponding pump passage for the transmission of the gas pressure variations thereto.

Similarly the drive means of FIGURE 10, which is preferred, could be used with the embodiments shown in FIGURES 1, 2, 3 and 12 by connecting the pipes **107** to a vertical cylinder, similar to **82** in FIGURE 11, which is partially filled with liquid and connected to the pump passages through liquid-filled pipes, as in the FIGURE 11 arrangement.

The mechanical drive means of FIGURE 10 could be coupled hydraulically to a hot liquid pump through a series of immiscible non-reactive liquids, with appropriate graded melting and boiling points, across which the temperature is progressively lowered to a value suitable for a solid moving piston.

For pumping liquids two other possible means of driving pumps similar to those shown in and described with reference to any one of FIGURES 4, 5, 6, 12 may be employed. Both of these involve the modification that the upper ends of the various passages be closed and incorporate the provision of injectors and/or valves in the upper parts of the passages. Both of them too, in operation, utilize thermal expansion of a gas or vapour in the upper parts of the passages to provide the driving forces on the liquid interfaces in the passages during blow strokes, so that the closed ends of the passages in effect provide chambers of the nature of internal combustion chambers or thermal expansion chambers, in which the pump liquid in the passage acts in effect as a piston, both operating on 2-stroke cycles, for which the drive and exhaust stages will hereafter be referred to as the blow and suck strokes.

Thus one such means, applicable to the pumping of both hot and cold liquids, employs the provision of a fuel injector and inlet and exhaust valves at the top of each passage. In operation, immediately prior to a blow stroke in each passage, when the liquid interface is at its highest level in the passage the inlet valve is opened and a burst of compressed air is blown in. Immediately following this combustible liquid fuel such as paraffin is injected as a spray, the inlet valve is closed and the fuel-air mixture is ignited and expands, either by spark ignition from a sparking plug provided near the top of the passage and arranged to spark at that moment or, in the case of a high-temperature liquid being pumped, by ignition at the high-temperature liquid interface. This expansion by combustion forces the liquid interface down the passage, thus providing the blow stroke. At the termination of the blow stroke the exhaust valve is opened to an external region maintained at some suitable pressure less than that prevailing in the liquid in the central channel whereupon the high pressure prevailing in the central channel forces the liquid interface upwards, almost to the top of the passage. This constitutes the suck stroke and almost empties the passage of burnt gases. The exhaust valve is then closed thus checking the rise and completing the suck stroke. This procedure is then repeated continuously in each passage, the required phasing of the pulsations in the passages being obtained by a suitable time sequence of operation of the valves and air/fuel injection stages for the passages.

The other drive means aforesaid is applicable only to the pumping of hot liquids. such for example as molten seed-slag mixture at over 1000° C. in M.H.D. generator plant or such as molten metal and employs the provision of an exhaust valve at the top of each passage and an injector for a non-combustible vapourisable drive liquid (e.g., water) let into the side of the passage. In operation, immediately prior to a blow stroke in each passage, when the pump liquid interface is at its highest level in the passage, with the exhaust valve closed and with the level of the interface of the hot pump liquid in the passage above the injector, a quantity of drive liquid is injected from the injector into the hot pump liquid. The injected drive liquid is almost immediately turned into bubbles of pressurised vapour. These rise rapidly through the hot pump liquid to the top of the passage, continuing to heat up and expanding in transit and an expansion takes place which forces the pump liquid interface down the passage, thus providing the blow stroke. When the pump liquid interface is at its lowermost desired level in the passage the blow stroke is terminated by opening the exhaust valve at the top of the passage to an external region maintained at some suitable pressure less than that pevailing in the pump liquid in the central channel. The high pressure prevailing in the hot pump liquid in the central channel then forces the hot pump liquid interface in the passage upwards almost to the top of the passage and clears the passage of spent drive vapour. The exhaust valve is then caused to close to check this rise and to complete the suck stroke. This procedure is then continuously repeated in each passage during the operation of the pump with suitable phasing of the strokes in the different passages.

In either of the two last described forms of driving means, in the phasing of the blow strokes in the various passages relative to one another various degrees of coupling between blow and suck strokes in different passages may be made in accordance with the patterns of liquid flows developed. Thus the blow stroke in one passage may be arranged to help the suck stroke in another and so that when 180° of cycle have elapsed the help may be reversed. This may result in faster suck strokes in each passage and a faster rate of pumping of the liquid along the central channel than would otherwise be obtained.

The pumps shown in FIGURES 1, 2, 3 and 12 may be used for pumping gases as well as liquids; thus corrosive gas could be pumped by means of a liquid/gas piston in reverse. For example, for each pump passage a mechanical piston such as **105** may be arranged to drive a liquid into and out of the bottom of a cylinder similar to **82** to the top of which cylinder the pump passage is connected. Very hot (but non-corrosive) gases may alternatively be pumped by a direct connection from a drive cylinder pipe such as **107** to a pump passage such as **11** in such a way that there is a temperature gradient along the connecting pipe.

It will be appreciated that any embodiment of the invention as described may contain any suitable number of pump passages similar to those illustrated, according to the pumping action required.

We claim:

1. A pump for fluid comprising a channel from which there project a plurality of passages spaced in succession along the length of the channel, the channel extending from a first region to a second region and the passages being connected to means entirely external of said channel for effecting pulsations of fluid along the passages when, in use of the pump, the channel is filled with fluid which extends into the passages, said pulsations being effected by the exertion of fluid pressure upon said fluid which extends into said passages, said means being arranged to so phase the pulsations between the various passages, and the passage inlets being so directed along the length of the channel, as to induce a net forward flow of fluid along the channel from said first to said second region when these regions are respectively at low and high pressures.

2. A pump according to claim 1, wherein the pulsations of fluid along the passages are produced by pressure variations in a gas linkage driven by external drive means.

3. A pump according to claim 1, wherein the pulsations of fluid along the passages are produced by pulsations of a liquid in a liquid linkage driven by external drive means.

4. A pump according to claim 1, wherein the passages and their entries into the channel are so oriented and shaped relative to the shape and direction of the channel that there is a net forward flow of fluid induced along the channel during both the suck and the blow strokes of the pulsations in any passage.

5. A pump according to claim 1, wherein the channel is of approximately cylindrical shape and the passages are in the form of tubes extending from the walls of the channel and inclined backwards with respect to the direction of flow of the fluid along the channel.

6. A pump according to claim 1, wherein the channel is of the form of a series of venturi parts and the passages extend as tubes through the falls to the center of the channel and within the channel are bent in such a way that their nozzles point along the axis of the channel in the desired direction of flow of fluid along the channel.

7. A pump according to claim 6 for pumping liquids downwards, wherein the said channel is vertical, each venturi wall in the diverging portion of its venturi diverges at a small angle to the axis of the channel and wherein a pair of passage tubes, each tube being cylindrical in its upper part but being tapered towards its nozzle at a small angle to its own axis, projects, each tube from opposite sides, through the walls of the channel into the region adjacent to and upstream of each of the venturi restrictions, and which pump is arranged to be operated with the pulsations in the passage tubes on one side of the channel approximately in phase with one another and approximately 180° out of phase with the pulsations in the passage tubes on the opposite side of the channel, the pressure responsive medium-liquid interfaces in each passage tube being arranged to oscillate substantially within the wholly cylindrical portion of it.

8. A pump according to claim 1, wherein the channel is vertical and each passage of said channel is approximately funnel-shaped, the aperture of each passage where it joins the channel extending round the circumference of the channel.

9. A pump according to claim 8 for pumping liquids downwards, wherein the passage funnels point downwards and the surfaces of each funnel passage are ridged internally in spirals about the axis of the central channel in such manner as to cause the liquid in each passage on a blow stroke to swirl into a vortex motion within the passage, and thereby upon ejection into the channel by virtue of the centrifugal force acquired, to be flung outwards and down the channel rather than upwards.

10. A pump according to claim 8 for pumping liquids upwards, wherein each of said passages has approximately the shape of an upwardly pointing funnel bent back upwardly upon itself to provide a bend at the bottom communicating between downward and upward funnel passage portions and wherein said drive means is adapted to effect pulsations in which the pressure responsive medium-fluid interfaces oscillate wholly within the downward portions of the passages so that no pressure responsive medium is blown out into the central channel and so lost.

11. A pump according to claim 10, wherein the surfaces of each funnel passage are ridged internally in spirals about the axis of the central channel in such manner as to cause the liquid in each passage on a blow stroke to swirl into a vortex motion within the passage, and thereby upon ejection into the channel to be flung outwards and up the channel rather than downwards.

12. A pump according to claim 9, wherein said spirals in each funnel passage are in the opposite clockwise sense to those in the passages immediately above and below it, thereby promoting by angular momentum coupling more efficient transfer of liquid out of any one passage upon a blow stroke and into the next in the desired direction of pumping.

13. A pump according to claim 8 for pumping liquids, wherein the axis of each passage portion adjacent to the central channel is inclined acutely to the axis of the central channel in the direction of pumping and the edge of the nozzle therebetween these two directions is sharp so that on a suck stroke the non-reciprocity of flow of a liquid forced around a sharp bend is utilized, liquid drawn into any passage being effectively lodge there until blown-out upon the coming of the next blow stroke.

14. A pump according to claim 1 adapted for the pump-

ing of hot molten slag-seed mixture in a combined heat exchanger and seed recoverer of M.H.D. generator plant and wherein the material of the walls of the channel and the passages is a suitable refractory substantially inert to the liquid being pumped.

15. A pump according to claim **1** designed for pumping liquids and in which the ends of the passages are closed, provided with inlet and exhaust valves or injectors, and associated with means for introducing drive fuel or vapourisable liquid so that the ends of the passages provide internal combustion chambers or thermal expansion chambers in which the pump liquid acts in effect as a piston arranged to be driven by the ignition of said fuel or expansion of said vapourisable liquid.

16. A pump according to claim **11**, wherein said spirals in each funnel passage are in the opposite clockwise sense

to those in the passages immediately above and below it, thereby promoting by angular momentum coupling more efficient transfer of liquid out of any one passage upon a blow stroke and into the next in the desired direction of pumping.

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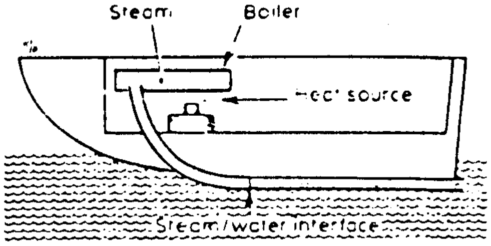
Steam-ships make a come-back

A new marine engine based on a toy that was popular 20 years ago is under development in the United States. The new engine is driven by steam and although its efficiency is similar to conventional steam engines, it has virtually no moving parts, is simple to construct, and poses far less problems of siting, particularly in small boats.



The toy ship on which the new engine is based is shown in the diagram. The boiler is filled with water which is heated to generate steam that forces water trapped in the pipe toward the stern. Once the water has evaporated from the boiler, negligible heat is transferred to the steam water interface, and residual steam in the pipe condenses, causing the pressure to drop: as a result water is drawn into the boiler, thus preparing it for the next cycle in which useful work is done.

In the toy, the pulse rate (that is, water in to water out) is a few cycles a second. The oscillation of the system produces a net propulsive force because the non-working stroke has negligible thrust.

Attempts were made some years ago to scale up the system but little progress was made because proper operation depends critically on the size of the various components. According to Peter Payne, of Payne Inc. in the US, which is now developing various forms of the engine for practical use, this failure was due to the fact that experimenters couldn't make a large unit work as a result of instability at the steam water interface. Beyond a certain diameter of pipe, the water wall doesn't remain "solid" under the action of the steam jet, but falls to the lower portion only of the tube, allowing steam to escape, along the upper portion, without much force for propulsive power.

To overcome this problem, Payne says that the jet action of the steam must be so fast as to prevent the water/steam interface from collapsing. He does this by making the boiler walls very thick, so that they absorb most of the energy from the heat source (a propane burner) when the boiler is empty. On the non-working stroke when cold water flows into the boiler it is suddenly heated, exploding into steam giving the necessary rate of discharge.

Another modification Payne has made is to split the inlet and outlet functions between two pipes. The inlet pipe points forward and has a non return-valve in it, admitting water only after the working stroke has been completed. With this arrangement Payne has reached a speed of 0·3 m/s (less than one mile/hr) in a 4·5 m aluminium rowboat, carrying a payload of 250 kg. Hardly world-shaking—but promising.

New Scientist. October 10, 1974.

S. A. E. Transactions. 1970, p. 346-352.
700095

A Pressure Exchanger For Marine Propulsion

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ABSTRACT

The turbopropeller transfers mechanical energy from the flow of a working fluid to that of a momentum medium. The same transfer can be effected more directly by pressure exchange, that is, through the work of mutually exerted pressure forces at appropriately generated interfaces between the two flows. The main drawback of pressure exchangers is that they are of necessity nonsteady-flow machines, highly time-sensitive and not easily controlled. This difficulty is overcome in the "bladeless propeller," a pressure exchanger whose flow processes, although nonsteady in the frame of reference in which they are utilized, admit a frame of reference in which they are steady. Available theories and experimental results indicate that this device promises to combine an attractive efficiency with advantages of compactness and mechanical simplicity.

THE SEARCH FOR NEW MODES OF MARINE propulsion has received increasing attention in recent years. Among the schemes that have emerged from this search are the water-jet (1)*, the underwater ramjet or gas-powered hydroduct (2, (2, 3), the underwater rocket (4), and the "cryptosteady pressure exchanger" or "bladeless propeller" (5-12). The available theories, and much of the experimental evidence that has been obtained so far, appear to indicate that the latter scheme offers the best promise of efficiency over a wide range of operating depths and speeds, as well as attractive advantages of ruggedness and compactness.

The bladeless propeller may be viewed as combining the functions of a turbine and a propeller, in that it transfers energy from a high-energy "primary" flow to a low-energy "secondary" flow for the purpose of thrust augmentation; that is, for the purpose of producing a higher thrust than could be produced by a directed jet of the primary flow alone. Furthermore, like the ejector pump, the bladeless propeller is a thrust augmenter belonging to that class of flow induction devices in which the transfer of mechanical energy from the driving primary to the induced secondary flow is direct, in the sense that it takes place through the work of mutually exerted forces at the interfaces between the two flows. However, in contrast to the conventional steady-flow ejector, where the energy transfer is effected through the work of shear forces, the bladeless propeller is a pressure exchanger (13): a device in which the transfer is effected in whole or in part through the work of interface pressure forces. Like all pressure exchangers, the bladeless propeller is of necessity a nonsteady-flow device, because no work is done by pressure forces acting on a stationary interface.

What is noteworthy in a comparison of these flow induction devices is that nonsteady-flow thrust augmenters, such as the pulsating-flow ejector, are capable of higher energy transfer efficiencies than their steady-flow counterparts, with generally reduced interaction lengths (14, 15). This fact is readily explained by the equations governing the time rate of change of total head or specific stagnation enthalpy of a particle. From the definitions of these quantities and the dynamical equation of motion one obtains for a fluid element, in the absence of body forces,

$$\frac{DH}{Dt} = \frac{\partial p}{\partial t} + \bar{u}\quad\bar{f} \tag{1}$$

if the fluid is incompressible, and

$$\frac{Dh^{\circ}}{Dt} = \frac{1}{\rho}\frac{\partial p}{\partial t} + T\frac{Ds}{Dt} + \frac{1}{\rho}\bar{u}\quad\bar{f} \tag{2}$$

if the fluid is compressible. These equations show that, in the absence of body forces, a reversible transfer of mechanical energy within a flow system is possible only in regions where the local derivatives $\partial p/\partial t$ are not zero, that is, in regions of nonsteady flow. From this it must also be concluded that a transfer of mechanical energy from one flow to another can be nondissipative only if and where both flows are nonsteady. This conclusion explains, and is supported by, the noted performance superiority of nonsteady over steady-flow induction devices.

OBSTACLES TO SUCCESS

The most serious obstacle to the successful development of nonsteady-flow thrust augmenters lies in the critical dependence of most of these devices on the accurate timing of moving mechanical parts to the wave and transient flow process on which their operation is based. Another obstacle is the laboriousness of nonsteady-flow analyses.

These difficulties are largely eliminated when one deals, instead, with "cryptosteady" processes (processes that are nonsteady but admit a frame of reference in which they are steady). Cryptosteady flows and interactions can be generated, analyzed, and controlled as steady-flow processes in this unique frame of reference, while retaining the potential advantages of nonsteady-flow processes in the frame of reference in which they are utilized.

The operating principle and characteristics of the cryptosteady pressure exchanger are briefly reviewed in this paper, with particular attention to the theoretical and experimental information that relates most directly to its application as a thrust augmenter.

CRYPTOSTEADY FLOWS AND INTERACTIONS

Consider the pressure field associated with a nonuniform flow system which is steady in a frame of observation F_s. At any point fixed in space relative to F_s the pressure is, of course, invariant with respect to time. On the other hand, at any point fixed in space relative to another frame of observation F that is moving at a velocity $\bar{V}$ relative to F_s, the pressure varies with time according to $\partial p/\partial t = \bar{V} \cdot \nabla p$. This means that, except where ∇p is zero or normal to $\bar{V}$, the flow system in F is nonsteady and undergoes energy exchanges that are absent in F_s. Since changes of the frame of observation are reversible, these energy exchanges are essentially nondissipative. Because of the existence of a frame F_s in which the flow is steady, the flow in F is cryptosteady.

Fig. 1 illustrates a simple cryptosteady pressure exchange situation. Here two flows deflect each other to a common orientation in a frame of reference F_s in which they are both steady. Neglecting energy exchanges by transport processes, which are slow compared to the processes considered here, no energy is exchanged between the two flows in F_s. Energy will, however, be transferred from one flow to the other in the frame of reference F of an observer 0' moving at an arbitrary velocity $\bar{V}$ relative to F_s, as evidenced by—and calculable from—the changing magnitudes of the velocity vectors in F. The energy so transferred is equal to the work done by the pressure forces which the interacting flows exert on one another at their interface. This work is zero in F_s, where the interface is stationary, but not in F, where the interface moves.

In practice, the simplest way to maintain the motion of F_s relative to F is to rotate the former at a constant angular velocity relative to the latter. Two arrangements of this type are shown in Figs. 2 and 3. The primary fluid is made to issue into the interaction space through slanted orifices on the periphery of a rotating member which is driven by the reaction of the issuing jets themselves. In the near field around the rotor, the primary fluid occupies spiral or helical regions that rotate, as if in solid-body rotation, about the same axis and at the same angular velocity as the rotor. The boundaries of these regions are the interfaces separating the primary from the surrounding (secondary) fluid, and their relation to the two flows is therefore substantially the same as that of blade or vane surfaces of the same shape, rotating at the same angular velocity.

Thus, the interacting flows exchange energy in a stationary frame by a mechanism which is very nearly the same as that of a turbopump or turbofan: the driving fluid forms a cascade of "pseudoblades," the action of which on the driven fluid is

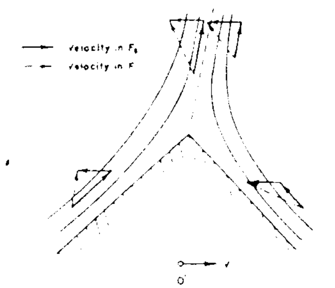

Fig. 1 - Schematic of cryptosteady interaction

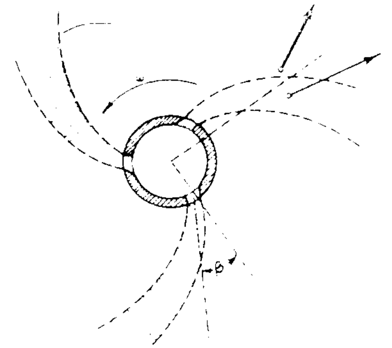

Fig. 2 - Radial-flow interaction

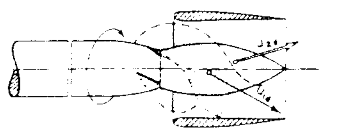

Fig. 3 - Axial-flow interaction

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*Numbers in parentheses designate References at end of paper somewhat similar to that of an impeller, while the effect of the interface pressure forces on the primary flow is essentially the same as the energy extracting action of a turbine. The energy acquired by the secondary fluid in this process is extracted entirely or partially from the primary flow, depending on whether the rotation of the rotor is produced and maintained solely by the reaction of the issuing jets or by other means. Fig. 4 shows the rotor of one of the radial-flow pressure exchangers that have been tested at RPI. Fig. 5 shows an axial-flow model. Fig. 6 is a flash photograph of pseudoblades produced by the latter model. Here water was used as the primary fluid, for the purpose of visualization. The flow of water is parallel to the interfaces in the frame of reference of the rotor but not in the laboratory frame of reference, as shown by the streaks, which are primary path lines in the latter frame of reference.

The mutual deflection that takes place when the two flows first come in contact with one another is normally followed by other forms of interaction. A second phase of pressure exchange takes place if the two flows are subjected together to Coriolis or other accelerations normal to their interfaces, as in a rotating flow field or in a passage of varying cross section. Transport processes also start simultaneously with the start of the pressure exchange phases but can be expected to proceed at a considerably slower rate. In the simplest analyses, mixing is assumed to start only after the pressure exchange phases are completed. **The portion of the mechanical energy transfer that takes place by pressure exchange approaches 100% of the total as the velocity of F_s relative to F is increased indefinitely. When this velocity is zero, no energy is transferred by pressure exchange and the device is reduced to a steady-flow ejector. In all other cases the efficiency of the bladeless pump or propeller is higher than that of the ejector, by virtue of the nondissipative pressure exchange phase.**

It has already been noted that, in every frame of reference except F_s, the velocities of the deflected flows have different orientations on the two sides of their interfaces. This fact may be utilized to separate the two flows after their nondissipative interaction, before mixing between them has made much progress (5). If the exit from the interaction zone is made up of two sets of stator vane passages, one with the orientation of $\bar{u}_{1k}$ and the other with the orientation of $\bar{u}_{2k}$, then a predominant portion of each of the two fluids will flow out through that set of passages which matches the orientation of its motion. Because of the steadiness of the flow pattern in F_k, the separation can also be effected by means of outflow ports fixed in F_s.

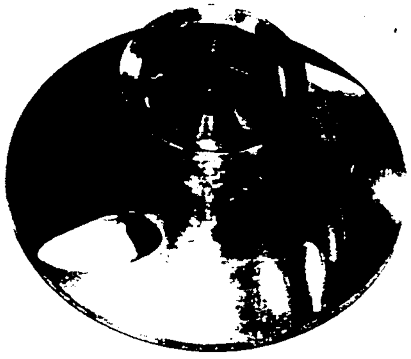

Fig. 4 - R.P.I. radial-flow test rotor

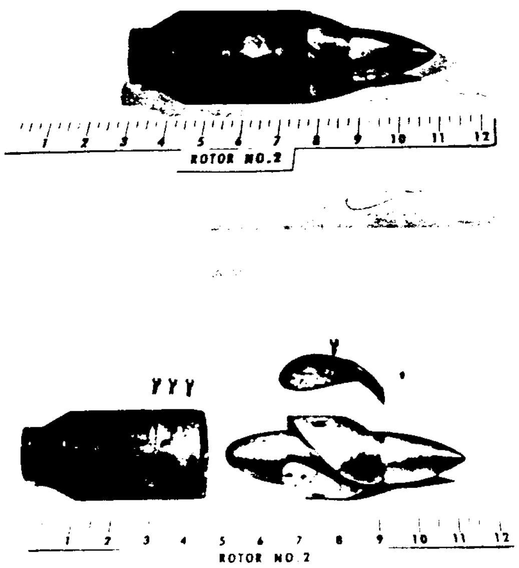

Fig. 5 - R.P.I. axial-flow test rotor

PERFORMANCE

Following Hohenemser (5), the angle between the primary discharge orientation in F_s and the local meridian plane will be referred to as the "spin angle," and the angle between the meridional projection of the primary discharge orientation and the rotor axis will be referred to as the "coning angle."

As noted, cryptosteady interactions are analyzed by studying the mutual deflection of the two flows to a common orientation in F_s (usually the rotor-fixed frame of reference) and then changing over to the laboratory frame of reference. In dealing with axial-flow situations of the type illustrated in Fig. 4, and if the width of the annular interaction space is small compared to its mean radius, the flows can be treated in good approximation as two-dimensional. Results of this type of analysis, for axial-flow interactions in ducts of constant cross-sectional area and for two values of the secondary-to-primary density ratio, are presented in Figs. 7 and 8. Here A_1 and A_2 denote the cross sectional areas of the primary and of the secondary flows, respectively, at the merger station, and Φ_0 is the thrust augmentation ratio in static operation. The analysis is for zero coning angle and does not account for mixing; that is, it assumes that the two flows deflect each other to a common orientation without mixing and that they

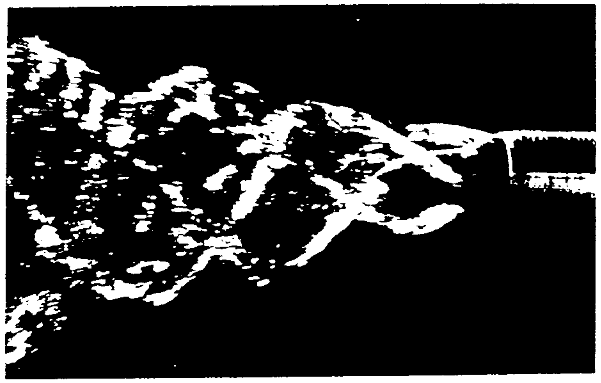

Fig. 6 - Water pseudoblades in air

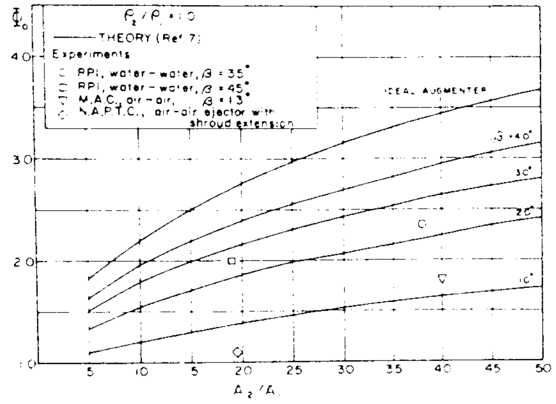

Fig. 7 - Static thrust augmentation of axial-flow model with $\rho_2/\rho_1 = 1.0$

are still unmixed when they leave the interaction space. Of the effects revealed by this analysis the most striking ones are those of the changes of spin angle and of secondary-to-primary density ratio. The effect of the latter changes is large and turns out to be of the opposite sign of what it is in the case of the ejector. By comparison, compressibility effects are very often found to be of little importance in this type of interaction.

The experimental results are seen to be in better agreement with the theoretical prediction when the spin angle is small than when it is large. One finds, of course, that because of the stipulated absence of mixing, the case of zero spin angle does not represent the ejector but rather a totally ineffective device for which the augmentation ratio is 1.0 throughout. It should be noted that the experimental point plotted in Fig. 7 for the ejector does not conform to the conditions of the analysis and of the other experiments, because it is for a model with a shroud extension for complete mixing, whereas the analysis precludes mixing and all other experimental points are for models whose interaction lengths were very short (approximately equal to the shroud diameter).

As is the case with most thrust augmenters, the thrust augmentation ratio of the cryptosteady pressure exchanger decreases with increasing forward speed. The effect, as predicted by the two-dimensional analysis for a water-water axial-flow model with a spin angle of 20 deg. and zero coning angle, and for an area ratio of 15, is plotted in Fig. 9.

An important improvement over the two-dimensional approach, for situations involving large area ratios or large spin angles, was introduced by Hohenemser in Ref. 8, with a thin-jet strip approach similar to the strip method of propeller theory. The primary jet is assumed in this approach to be thin and to penetrate the interaction space at a small coning angle (upper portion of Fig. 10). The interaction is then treated as an infinite succession of infinitesimal steps, in each of which the primary jet and the elemental layer of secondary fluid which it penetrates deflect each other to a common orientation in F_s. Coriolis and centripetal accelerations are again neglected, and no account is taken of mixing. The resulting axial velocity distribution within each flow at the shroud exit is nonuniform, and the calculated thrust augmentation ratio is lower than that predicted by the two-dimensional theory (Fig. 10). It is noteworthy that the strip theory predicts, for each area ratio, an optimum spin angle.

As can be seen from Fig. 10, the predictions of the strip theory are in remarkably good agreement with experimental results. Nevertheless, this theory must be considered too pessimistic as a basis for prediction of potential performance, because obviously, if the primary jet were wide enough to span the entire interaction space, the interaction would be more accurately described by the two-dimensional theory. The predictions of more refined future analyses will probably lie somewhere between the results of these two theories.

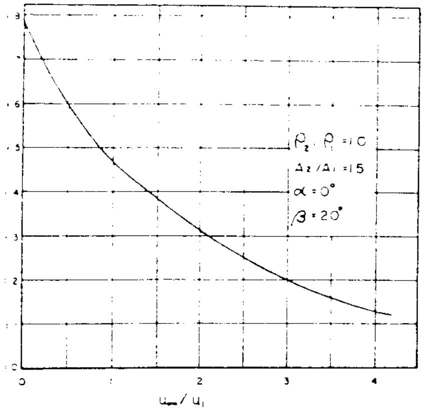

Fig. 9 - Effect of forward speed on thrust augmentation

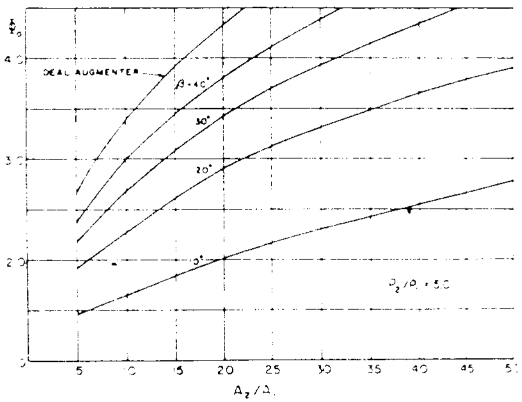

Fig. 8 - Static thrust augmentation of axial-flow model with with $\rho_2/\rho_1 = 5.0$

The effect of constant-area mixing following the deflection phase is shown in Fig. 11. The dashed line gives the performance as calculated by the two-dimensional approach without mixing, and the solid line the results obtained by Hohenemser (9) by the two-dimensional approach with a constant-area mixing phase following the deflection phase. The effect of this mixing may be favorable or unfavorable, depending on the spin angle.

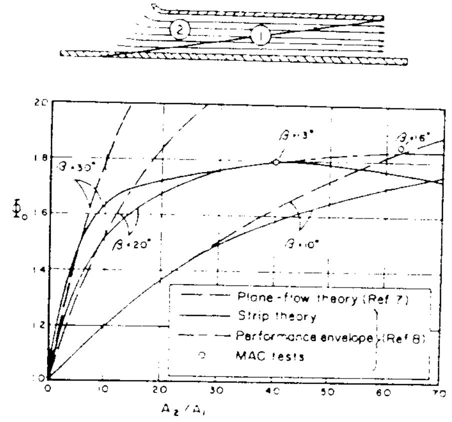

Fig. 10 - Static performance of bladeless propeller for $\rho_2/\rho_1 = 1.0$

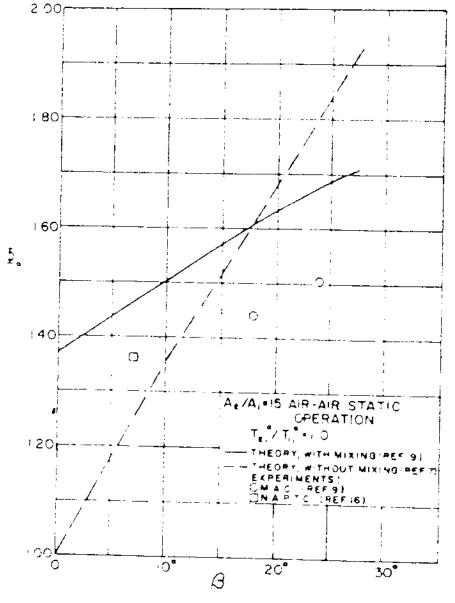

Fig. 11 - Static performance of bladeless propeller with and without constant-area mixing following the deflection phase

Coriolis and centripetal accelerations can no longer be neglected when the coning angle is large, as, for example, in the configuration shown in Fig. 12. These accelerations, first accounted for in the analyses of Refs. 10 and 11, can have large and beneficial effects on performance, as can be seen from Fig. 13. The experimental results reported in this figure were obtained by Avellone (17) with air-air interactions.

The effect of mixing during the deflection phase—always an adverse effect—has been studied by Hohenemser and Porter (12) through the use of an **"entrainment coefficient,"** which measures the rate of mass transfer from the secondary to the primary flow. Even with the entrainment coefficient corresponding to the highest mass transfer rate observed in stationary jet cascade experiments, the predicted performance loss is almost insignificant. The actual performance loss due to mixing during deflection is believed to be a good deal higher in the rotary-jet situation, where mixing is promoted by the curl component associated with the rotation of the rotor.

Steam-water interactions have recently been studied theoretically by Sarro and Kosson (18) and experimentally by Avellone and Sarro (19, 20, and 21). The Sarro-Kosson analysis is based on a strip approach and stipulates, in addition, that the two flows are separated after deflection and separately discharged. The conditions of the tests were probably intermediate between those of the Sarro-Kosson theory and those of the two-dimensional theory. However, direct comparison of

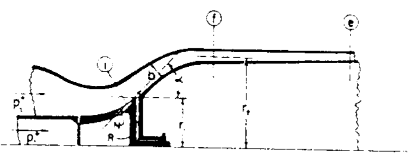

Fig. 12 - Configuration of acceleration with large coning angle

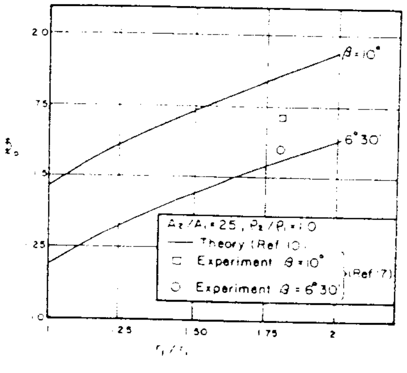

Fig. 13 - Static performance of bladeless propeller with large coning angle

the experimental results with the theoretical ones is not yet possible in this case because the primary fluid used in these tests was steam, whereas numerical results of the theories are available only for the case of a noncondensing primary. In all tests the maximum augmentation was obtained with very low spin angles (between 1 and 2 deg). Thrust augmentation ratios as high as 4.85 in static operation and 4.25 at 16 knots have recently been measured in the Grumman steam-water tests, with a spin angle of only one degree (20).

CONCLUDING REMARKS

Although relatively little effort has gone into the study of the bladeless propeller so far, the available evidence of its potential merit as a thrust generator is already impressive. A good deal more, however, remains to be done, particularly in relation to the possible applications of this device to marine propulsion. Means must be found to inhibit the dissipation of the primary jets in the deflection phase, and criteria must be developed for rotor and shroud contouring and for the selection of primary orifice shapes. Special attention should also be given to alternative ways of generating the pseudoblades, to the utilization of two-phase interactions, and to the possibility of **compounding cryptosteady pressure exchange with other** mechanisms of **energy transfer**

SYMBOLS

A = cross-sectional area

$\overline{f}$ = force per unit volume due to surface viscous stresses

h = specific enthalpy

p = static pressure

s = specific entropy

t = time

$\overline{u}$ = flow velocity

α = coning angle (inclination of meridional projection of primary discharge axis to rotor axis)

β = spin angle (inclination of issuing primary jet axis to meridional plane)

ρ = density

Φ = thrust augmentation

Subscripts

o = static operation

1 = primary flow

2 = secondary flow

∞ = free stream

d = deflected flow

i = merger station

Superscripts

o = stagnation

$'$ = flow conditions after isentropic expansion to free-stream static pressure

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700102

A Technique for Reduction of Axle Generated Noises

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IN THE PAST few years, the ambient noise level in the Cadillac vehicle has been substantially reduced due to noise and vibration advances in areas such as exhaust noise, engine noise, wind noise, road noise, and mechanical resonant booms. Axle noise improvements were desirable to further improve the noise quality of the car.

The purpose of this paper is to describe a different approach to an axle noise solution which Cadillac successfully completed for the 1968-1969 model year. The approach used is not concerned with work on axle manufacturing quality. It is concerned with identifying vibration resonances of chassis components and their associated transmission paths. Developing a means to alter them will then attenuate noise in the vehicle

Exciting forces for the resonances come from the meshing action of the axle gear set. With an axle of excessive force levels, several separate chassis component systems will be excited into noise producing resonances, each at their own frequency. Cadillac's resonant frequencies ranged 330-850 Hz.

The effect of the widely scattered resonances is to produce several audible and often objectionable axle noise periods throughout a wide speed range of the car (30-80 mph).

INVESTIGATION TECHNIQUE

The objective of the noise and vibration investigation is to establish a correlation between axle noise periods and chassis component response periods. This requires a systematic search of undercar components on a running car using vibration transducers, to reveal individual parts or systems that are resonant and, therefore, possibly causing the axle noises. Past Cadillac experience has shown that the rear suspension upper control arms play a part in producing some of the objectionable noise periods. Heavy 9-lb weights, when bolted into these arms, produce significant axle noise reductions in the 60-66 mph car speed range. Also, when lower spring rate upper control arm bushings (20% decrease) are installed, the loudest noise periods are reduced in intensity. It is felt that this is the result of increased isolation in the transmitting path. These partially successful changes indicate that after the resonant systems are defined accurately, retuning them will change the axle noise periods, hopefully, for the better.

The chassis dynamometer is one of the most important tools in investigating the axle noise problem. It consists basically of a large set of rolls upon which the driving wheels of the study car rest. The rolls are connected by a common shaft to a 250 hp electric dynamometer. The road loads due to wind resistance, rolling resistance, grades, and car inertia are closely duplicated, using electronic shaping circuits. This allows the operator of the vehicle on the rolls to drive through its speed range in the same manner as that done on the road. In this way, the axle noise periods heard on the road are easily duplicated in the study car on the chassis dynamometer. A picture of the car on the chassis dynamometer is shown in Fig. 1 Without the availability of this facility, the study of chassis responses would be very difficult. The floor of our chassis

ABSTRACT

Vehicle "axle noise" resulting from the resonant vibration of rear suspension and driveline components was measured and then reduced by changing their structural properties. No changes were made to the rear axle assembly to achieve the noise reductions.

Mesh frequency and twice mesh frequency response measurements of the rear control arms, axle housing, and propeller shaft correlated exactly with vehicle noise periods

Structural stiffness and mass changes to these components with the effect on noise and vibration response is presented.

A description of the eventual production hardware which attenuated "axle noise" to an acceptable level is also presented.

Record of the 10th Intersociety Energy Conversion Engineering Conference.
8/18-22/75, University of Delaware, Newark. p. 534-547.

it is possible to calculate ΔS for any segment of path along which T is constant and for which ΔQ is measured.

To see what is going on, suppose that the lines AB and CD are horizontal in Fig. 7c (that is, the short adiabatic expansion and contraction are negligible) and that the distances BB' and DD' are made to approach zero. Then the amount of energy available to do work is $\Delta Q_{AB} - \Delta Q_{CD}$ while the total heat input is ΔQ_{AB}. But

$$\Delta Q_{AB} = \int_A^B TdS = T_H \Delta S$$

$$\Delta Q_{CD} = \int_C^D TdS = T_C \Delta S \quad .$$

$$\text{So Maximum Efficiency} \equiv \varepsilon_c = \frac{\Delta Q_{AB} - \Delta Q_{CD}}{\Delta Q_{AB}} = \frac{T_H \Delta S - T_c \Delta S}{T_H \Delta S}$$

which is the Carnot efficiency.

The system we have described is an approximation to this. If we take into account the segments BB' and DD', we may calculate a better approximate efficiency thus:

$$\varepsilon \approx \frac{\text{energy available to do work}}{\text{heat energy input}}$$

$$= \frac{\left(\int_A^B T_H dS - \int_B^{B'} T_H dS\right) - \left(\int_C^D T_c dS - \int_D^{D'} T_c dS\right)}{\left(\int_A^B T_H dS - \int_B^{B'} T_H dS\right)}$$

$$= \frac{\left(\Delta Q_{AB} - \Delta Q_{BB'} - \Delta Q_{CD} - \Delta Q_{DD'}\right)}{\Delta Q_{AB} - \Delta Q_{BB'}}$$

Note that it is necessary to measure the above ΔQ's in order to estimate an efficiency. This has never been done systematically. Using crude data which are available (and a somewhat more complicated argument) Oleh Weres has estimated that such an engine may achieve up to 80% of Carnot efficiency.[5]

We should also try to estimate the change in temperature associated with the adiabatic changes in tension at A and C. At A, the wire is hot (fully transformed) and therefore the only heat change will be due to normal quasi-elastic deformation, i.e. the heat coefficient is a normal C_v. The isotherm is very steep, so that only a minor change in elongation takes place here. Hence the temperature change should not be a significant part of $T_H - T_C$.

Conversely at C, the wire is completely in its cold state and therefore has a normal C_v. If the wire is a trained wire so that its elongation is "automatic" and if T_C is sufficiently below the transition temperature range for the (minimal) tension, then the change in elongation in going from C to D should be a small part of the total, and again the temperature change should be small compared to $T_H - T_C$.

I have outlined in this paper the development of one class of Nitinol heat engines during the period of one year. In such a short span of time it is not possible to evaluate the probability of eventual success of these or similar engines. Considerable interest has been aroused in these engines because of their simplicity, novelty, and obvious timeliness. The prospect of utilizing waste heat energy is but one of the possibilities for alleviating our almost total dependence on fossil fuels. I am convinced that there is no one energy of the future: we should plan to develop alternative and competing means of energy conversion.

Where might Nitinol engines fit into this picture? If very large engines become practical, I should like to use a saline-gradient solar pond[7] to provide the heat source and use cold ocean bottom water as a heat sink. The converter should be an engine which can survive in very saline water, and this a natural environment for the very corrosion-resisting Nitinol alloy. I hasten to add that the economic feasibility of such a plan is very much in question. But this situation may change as fossil fuels become more dear.

Footnote and References

*This research was supported in part by the U.S. Energy Research and Development Administration and by a grant from the National Science Foundation.

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A PROGRESS REPORT ON PULSEJETS

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Introduction

At last year's Conference, the writer described the beginnings of a new engine cycle; the water pulsejet ship propulsor and its analogous water pump. The paper should logically have been assigned to the "Marine Power Systems" section of the conference, but was apparently deemed sufficiently far-out to end up in "New Engines and Novel Devices."

As is usual in a new technology, a retrospective reading a year later reveals severe shortcomings and errors in that earlier paper. But more than that, it discussed a very narrow part of a much wider field; that of pulsejet engines and pumps. In the present paper, we attempt to rectify this situation, for propulsive engines at least, by proceeding from the general to the particular, rather than the reverse

In order to do useful work, an engine needs to perform the following functions:

Raise the pressure of the working fluid.

Then add heat to it.

Then expand the fluid, doing useful work in the process.

Reject heat from the fluid when it is at a lower pressure or have access to a fresh change of working fluid; the steam engine is an example of the first; the internal combustion engine and the air ramjet are examples of the second alternative.

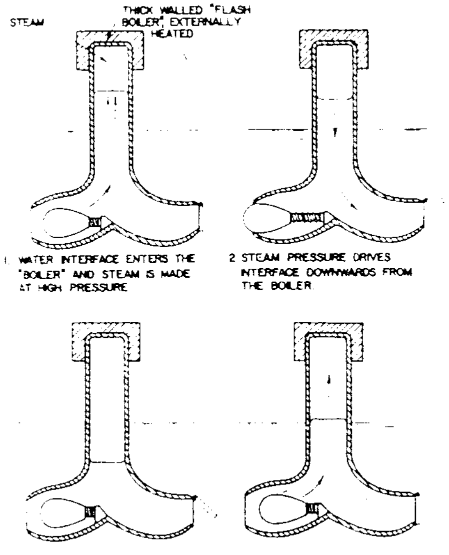
In order to efficiently convert heat to mechanical work, the "compression ratio" must be high; typically between 5:1 and 20:1 for an air-breathing engine. We usually achieve this compression by pistons or continuous flow "compressors" (axial or centrifugal) except in the special case of the ramjet.* An exception is the much misunderstood pulsejet family of engines. In a sense, these use a "piston" to raise the pressure of the working fluid, but the "piston" is the fluid itself, so that there is no need for any moving mechanical parts. The pulsejet can achieve very high compression ratios even under static thrust conditions; possibly higher pressures than are achievable in any other practical way. Thus, it offers the possibility of being an efficient engine, despite the fact that it can clearly be the lightest and the cheapest as well. This seems too good to be true. And why do we have no pulsejets now, or any work proceeding on them if they are so promising?

The answer must be a complex and fascinating illustration of what can happen in technologies which are not properly understood by workers involved in the developmental process; unfortunately, an attempt to analyze the reasons for this lack of basic research would require a separate paper in itself, which probably needs to be written by another author!

Steam Water Pulsejets (SWPJ)

The SWPJ was unequivocally invented by McHugh in 1916. At the time he was trying to invent something else, and it (not unnaturally) took him some time to realize precisely what he had accomplished. His engine could be built only in toy sizes because (we now know) it required surface tension to stabilize the interface between the steam and the water.

Hundreds of thousands of toy "putt-putt" boats were built and sold in the twenties and thirties. A great deal of money was wasted in attempting to scale these "putt-putt" engines up to larger sizes; wasted because (presumably) no one really knew how the engine worked, let alone had a good math model of its operation. Even if they had been given the equations of motion, their solution would have been unbearably tedious in that pre-computer age.



1. WATER INTERFACE ENTERS THE "BOILER" AND STEAM IS MADE AT HIGH PRESSURE.

2. STEAM PRESSURE DRIVES INTERFACE DOWNWARDS FROM THE BOILER.

3. THE STEAM CONDENSES ON THE COLD WALLS, AND THE LOW INTERNAL PRESSURE ARRESTS THE DOWNWARDS MOTION OF THE INTERFACE.

4. THE INTERFACE ACCELERATES TOWARD THE "BOILER" TO REPEAT THE CYCLE.

FIG. 1. A STEAM CYCLE WATER PULSEJET. THE EXIT VALVE (SHOWN DOTTED) IS HELPFUL IN VISUALIZING THE CYCLE, BUT IS UNNECESSARY IN PRACTICE.

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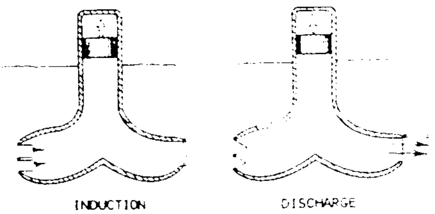
* The air breathing ramjet, as its name implies, obtains most of the ram-air pressure rise $\frac{1}{2}\rho u_0^2 [f(M)]$ by slowing the air down prior to combustion of fuel in it, subsequently expanding the air through a nozzle to obtain a positive thrust. It cannot produce a static thrust, its subsonic thrust is low and its specific fuel consumption high. The water ramjet is not a heat engine, but a propulsor which requires a supply of compressed gas.

Quite fortuitously, we were able to pick up the ball again in 1971, and as a result of modern developments in fluid mechanics and computers, were successful in breaking the size barrier. Some of our early results were described in last year's paper. Some of our later work is being funded by the Office of Naval Research, and we feel reasonably confident that the SWPJ will have a future in marine propulsion.

By way of recapitulation, the operation of a bifurcated steam pulsejet is indicated in Figure 1. The figure is self-explanatory, but it should be remarked that the "boiler" is nothing like a conventional boiler, and that the term may give rise to misconceptions. The steam water pulsejet (SWPJ) boiler is a compact block of metal - usually copper or brass - which has demonstrated heat transfer rates as high as 25,000 Btu/ft^{2}hr°F and 5 x 10^6 Btu/ft^{2}hr. These figures are orders of magnitude better than a conventional "pool boiling" or superheating boiler. Additionally, the steam energy is sufficient for only one stroke, so that conventional boiler safety hazards are not involved.

The Piston Pulsejet (PPJ)



This piston pulsejet (PPJ), sketched above* for a water propulsion application, may use any "piston engine cycle" to drive it, since the driving cycle is separated from the working fluid by a piston. So far we have only built one driven by compressed air (Figures 2 and 3) and paradoxically, it turns out that, for this particular motive power, the piston is not actually needed. The water itself acts as the piston. Relying on "Humphrey Gas Pump" experience, we anticipate that the same will be true for PPJ's operating on the Otto cycle

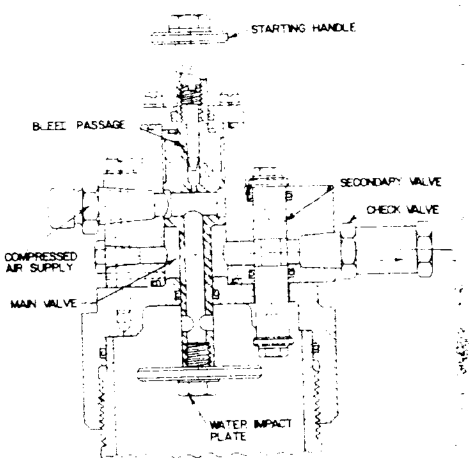


FIG. 2. CROSS-SECTION OF THE PAYNE, INC. THREE-INCH COMPRESSED AIR DRIVEN WATER PULSEJET.

Initially charged with water, downwards motion of the piston results in a pressure build-up which closes the intake valve and expels a propulsive jet to the rear. When the pressure above the piston has fallen to a sufficiently low value, the downwards motion is arrested and the piston starts to rise, allowing a fresh charge of water to enter the intake valve.

The water charge in the engine acts like a "linear flywheel"; analogous to the familiar rotating flywheel of

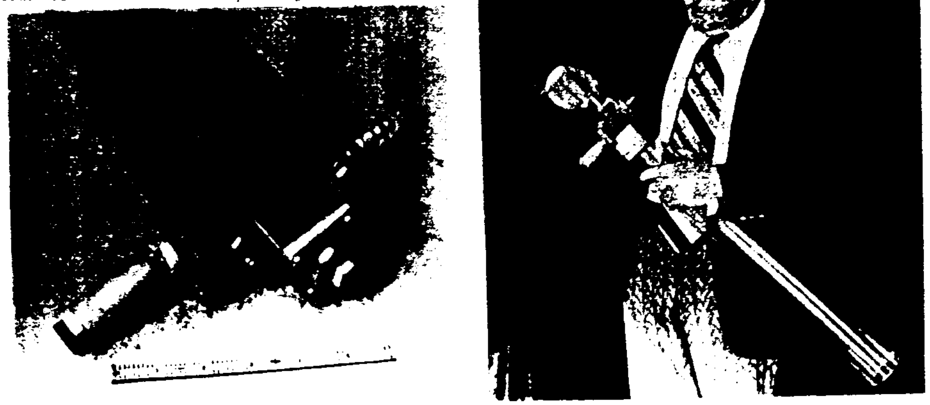
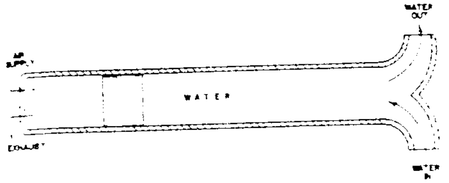


FIG. 3. TWO VIEWS OF THE PROTOTYPE PAYNE, INC. COMPRESSED AIR-DRIVEN WATER PULSEJET.

a rotary shaft piston engine. The piston is driven by pressure changes above it, like any conventional engine, and these pressure changes can be induced in any of the conventional ways, i.e.

Internal Combustion - Diesel cycle
- Otto cycle
Vapor - Rankine cycle
Gas Pressure Change - Stirling cycle
- Ericsson cycle
- Roesel cycle



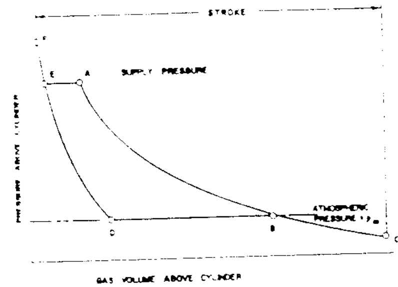


FIG. 4. P-V DIAGRAM FOR A COMPRESSED-AIR-POWERED PULSEJET.

The compressed-air-driven pulsejet of Figures 2 and 3 is the easiest to understand, and of course its performance is a basic analog to any of the above cycles. Referring to Figure 4, the cylinder is charged with compressed air between points E and A and the air valve closes at position A. The air continues to expand, and the downward velocity of the "piston" increases. At position B, the internal air pressure has fallen to ambient, and the piston starts to slow down, stopping and reversing its motion at C. It returns to B along the same adiabatic, at which point a valve opens to vent the cylinder to atmosphere until point D is reached, when the valve closes. The remaining air is compressed by the upward motion of the piston until it stops at point F and bounces back. The air supply valve opens at E, the space above the piston is again charged with compressed air, and the cycle repeats itself.

Such a cycle is modelled in Payne, Inc.'s computer program "AUPJS," which is based on the equations given in the last section of this paper, and some outputs for an engine of randomly selected geometry (Table 1) are given in Figures 5 through 14.

In Figures 5 through 9, the supply pressure is held at the fairly low value of 450 psi, while the ship speed and exhaust nozzle area are varied. Maximum propulsive efficiency (Figure 8) for the arbitrarily selected geometry is seen to be 0.62 at 80 ft/sec (47.4 knots) when the thrust level is 288 lb.

Table 1. Example Compressed-Air-Powered Pulsejet

Total length	8 ft
Diameter	6 in
Ram recovery efficiency	1.0
Vent valve closes at	7.9 ft from nozzle
Compressed air valve closes at	7.6 ft from nozzle
Supply pressure is	- 450 psi in Fig. 3-7 - Variable in Fig. 8-12
Nozzle area ratio* is	- Variable in Fig. 3-7 - 2.0 in Fig. 8-12
Ship velocity is	- Variable in Fig. 3-7 - 30 ft/sec in Fig. 8-12

* Nozzle area ratio is defined as being the ratio of "piston" area to water exhaust nozzle area

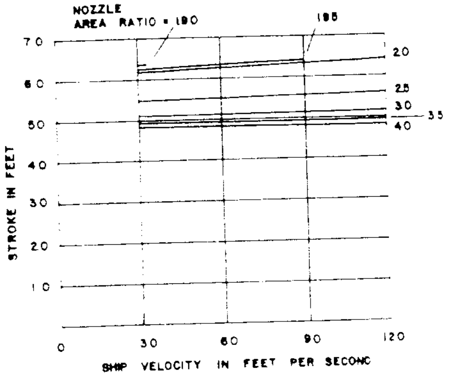


FIG. 5. PISTON STROKE FOR THE EXAMPLE PULSEJET.

In Figures 10 through 14, the nozzle area ratio was arbitrarily held constant at 2.0 and the ship speed at the relatively low value of 30 ft/sec, while the supply pressure was increased. As would be expected, the thrust increases rapidly with supply pressure, and the propulsive efficiency falls. At maximum pressure, the high thrust level of 1300 lb is quite remarkable for a six-inch-diameter engine.

These calculations assume a relatively sophisticated valving system, where the location of the piston (or water interface, if the engine has no piston) is sensed in some way and appropriate valves are opened and closed.

A simpler practical embodiment is that shown in Figures 2 and 3. In this engine, the rising water interface strikes the impact plate and pushes the main valve to the "up" position, allowing compressed air to flow into the space above the water. The water is pushed down by the air pressure, the main valve falls to a closed position because the force associated with the pressure above its bleed passage becomes greater than the force tending to hold it open. After it falls, it opens the cylinder to the secondary valve passage, but the secondary valve itself is held closed by the overpressure within the cylinder. As the air continues to expand m-

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* The exit valve (shown dotted) is helpful in visualizing the cycle, but unnecessary in practice

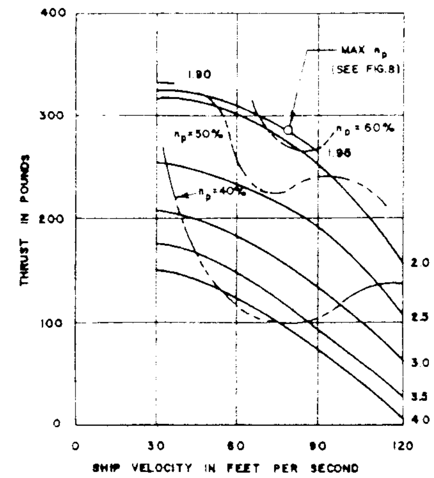


FIG. 6. AVERAGE THRUST FOR THE EXAMPLE PULSEJET.

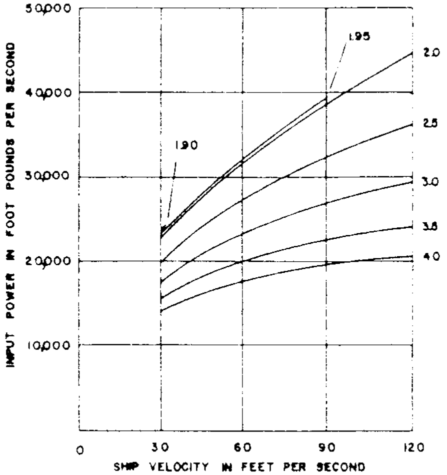


FIG. 7. AIR POWER REQUIRED BY THE EXAMPLE PULSEJET.

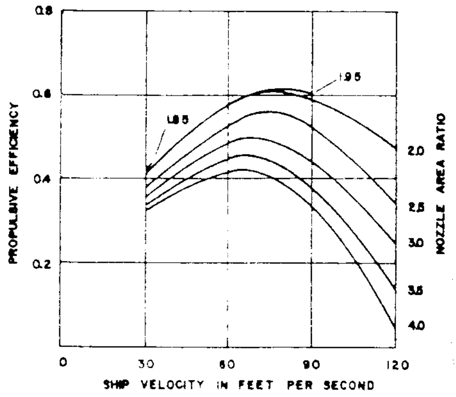


FIG. 8. PROPULSIVE EFFICIENCY OF THE EXAMPLE PULSEJET.

NOTE: PROPULSIVE EFFICIENCY IS DEFINED BY

$$\eta_p = \frac{\text{THRUST} \times \text{SHIP VELOCITY}}{\text{INPUT POWER OF COMPRESSED AIR}}$$

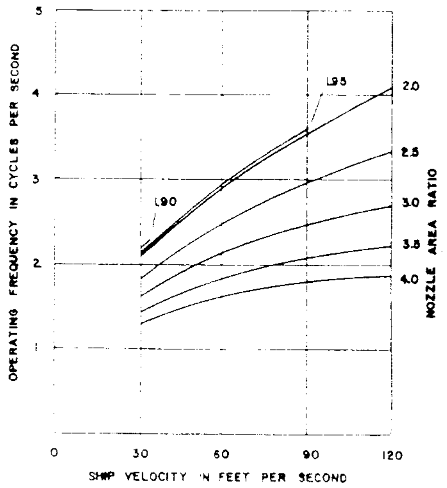


FIG. 9. OPERATING FREQUENCY OF THE EXAMPLE PULSEJET.

pushing the water down, its pressure falls to ambient and the secondary valve falls to an open position. As pressure drops further, the check valve closes, preventing outside air from entering the cylinder.

The further reduction of internal pressure finally arrests the water motion and starts to accelerate it upwards. When the internal pressure has risen to ambient, the check valve opens and allows the cylinder air to be vented to atmosphere, as the interface continues to rise. Finally, the interface strikes the water impact plate and the main valve is forced to the "up" position, allowing a fresh charge of air to enter the cylinder, the pressure of which closes the secondary valve and starts to accelerate the water downwards, to repeat the cycle. The "lower end" of this engine is illustrated in Figure 15.

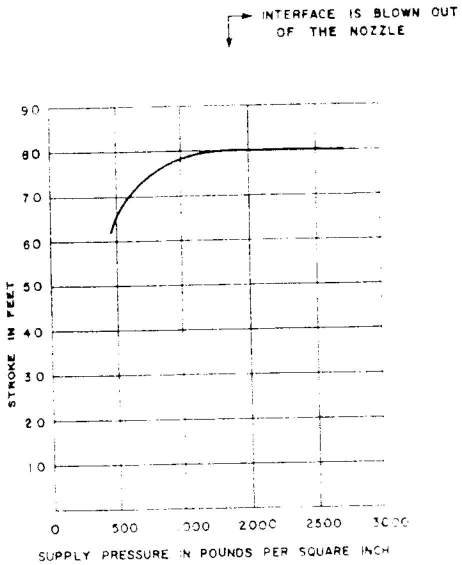


FIG. 10. EFFECT OF INCREASING SUPPLY PRESSURE ON THE OPERATING STROKE OF THE EXAMPLE PULSEJET.
NOZZLE AREA RATIO = 2.0
SHIP VELOCITY = 30 FT/SEC

Conceptually, the same engine can be converted to a diesel cycle by using the double piston configuration sketched in Figure 16. The space between the two pistons is used to induct and compress free air. Conceptually, such an engine would work very similarly to the compressed air cycle, except that part of the compressed air would purge the cylinder before the piston closed off the exhaust port "D", and that the high pressure is provided by compression ignition of fuel in the air rather than as gas from a reservoir.

Figure 16 shows a separate compressed air reservoir in order to tie in to the preceding discussion of a compressed air pulsejet. In practice, of course, the air would pass directly from check valve "C" to the cylinder. An already established example of such a configuration is familiar in the diesel-powered pile drivers which are in extensive use.

The Air Breathing Pulsejet (ABPJ)

If the pun will be forgiven, this engine started its operational life with a bang. Probably the most successful ones ever built were the first ones, used in the German V-1 flying bombs (Figure 17a). After the war, versions were developed to power gliders, as rotor-blade tip drive engines for helicopters and (perhaps the most successful) the Dynajet engine for model airplanes. There was a trend in this country to minimize the tailpipe length giving operating frequencies around 100 Hertz, vis a vis 60 Hertz for the V-1, which resulted in lower compression ratios and hence lower efficiency. No one really understood how the engine worked, as witness the English engineer who, tired of mending valve failures, closed up the front end entirely with a steel plate, and found that

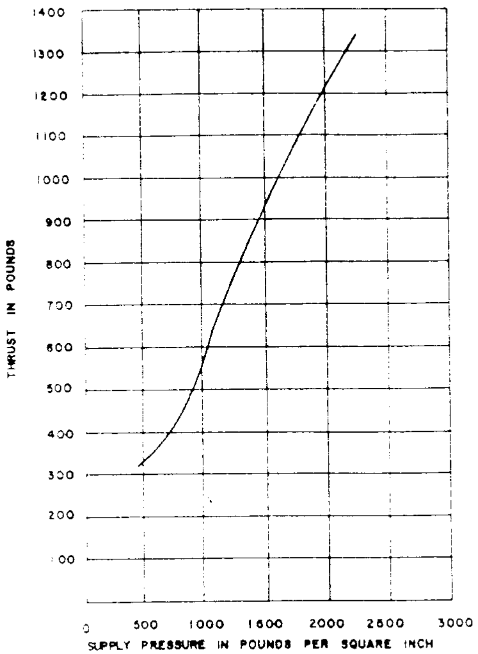
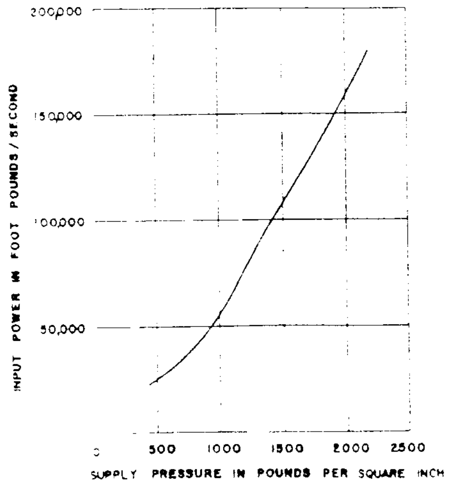


FIG. 11. EFFECT OF SUPPLY PRESSURE ON AVERAGE THRUST.
NOZZLE AREA RATIO = 2.0
SHIP VELOCITY = 30 FT/SEC



[illegible] EFFECT OF SUPPLY PRESSURE ON INPUT POWER REQUIRED, WHEN THE INTERFACE IS ALLOWED TO BLOW OUT OF THE NOZZLE. NOZZLE AREA RATIO = 2.0.
SHIP VELOCITY = 30 FT/SEC

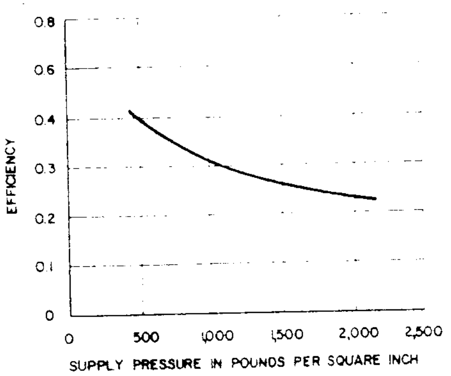


FIG. 13. EFFECT OF SUPPLY PRESSURE ON PROPULSIVE EFFICIENCY WHEN THE INTERFACE IS ALLOWED TO BLOW OUT OF THE NOZZLE.
NOZZLE AREA RATIO = 2.0
SHIP VELOCITY = 30 FT/SEC

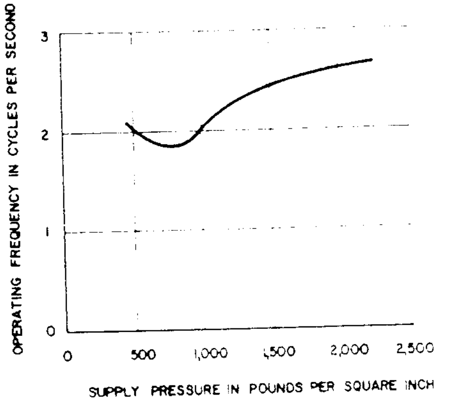


FIG. 14. EFFECT OF SUPPLY PRESSURE ON OPERATING FREQUENCY OF THE EXAMPLE PULSEJET
NOZZLE AREA RATIO = 2.0
SHIP VELOCITY = 30 FT/SEC

his engine ran just as well. He failed to follow through on this discovery, for reasons now unknown. The French discovered that the "aerodynamic valves" of Figure 17b could be made to work rather well, but that significant "negative thrust" pulses were coming out of the "intake valve." So with fine Gallic logic, they bent their engine double, so that both "inlet" and "exhaust" were pointing aft. They too failed to perceive what their engine was trying to tell them that the "intake" wasn't necessary at all.

Based on our recent work with water pulsejets, we think that the ideal low speed configuration is probably that of Figure 17c, just a tube with a closed end. It could perhaps be used as the motive power for a subsonic ramjet as shown, for example, in Figure 18. Or it could be bifurcated (like the water pulsejets) as shown in

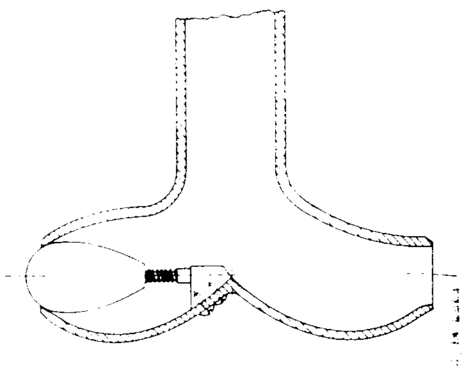


FIG. 15. THE LOWER END OF THE THREE-INCH COMPRESSED AIR DRIVEN PULSEJET, SHOWN IN FIGURES 1 AND 2.

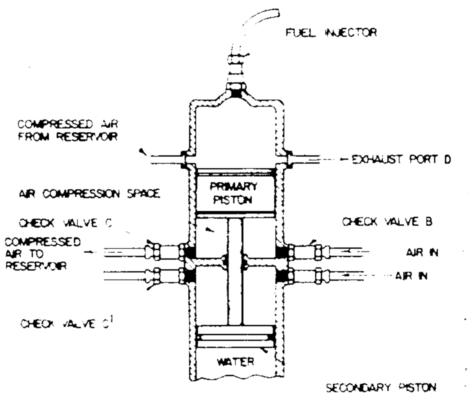


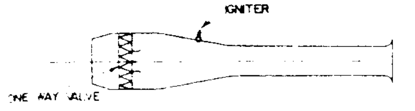
FIG. 16. A DOUBLE-PISTON DIESEL PULSEJET.

Figure 19. Note that this separation of "inlet" and "exhaust" opens a new dimension in permissible geometric variations, and that if our water pulsejet work is anything to go on, one would expect to achieve performance improvements when the exhaust is constricted.

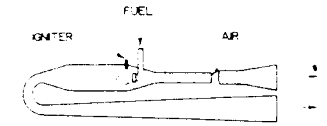
The Ideal Propulsive Efficiency of a Pulsejet

Modern propulsors are more or less steady state, continuous flow devices, even though a detailed analysis reveals some small scale velocity variation in their wakes, due to vorticity and viscous effects.

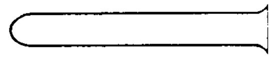
In contrast, man's earliest mechanical propulsors -- paddles, oars and sweeps -- produced a pulsed wake. These were all man-powered. In ships, the transition to steam power was accomplished with the paddle wheel, itself a transition between discrete pulses in the water and the quasi-continuous flow of the modern propulsor.



(a) THE "STANDARD" ABPJ CONFIGURATION (60-100 HERTZ) (1940)



(b) THE FRENCH SNECMA "ECREVISSE" VALVELESS PULSEJET (1950)



(c) LOW SPEED PAYNE CONFIGURATION (1974)

FIG. 17. THREE ABPJ CONFIGURATIONS.

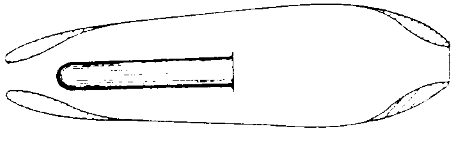


FIG. 18. AN AIR-BREATHING PULSEJET/RAMJET COMBINATION.

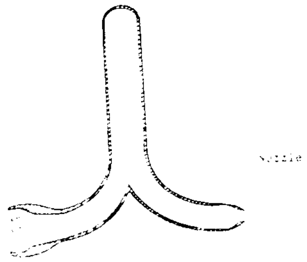


FIG. 19. A HIGH-SPEED AIR-BREATHING PULSEJET SUITABLE FOR SUBMERGED WING OR FIN INSTALLATION.

Historically, the builders of engine powered ships have adapted engines already developed, usually with a time lag of a decade or two after they were first introduced into land-based or aircraft use.* The power output was almost invariably available from a rotating shaft, and the simplest course was therefore to connect this shaft to a rotary water propulsor, usually through an intermediary gearbox.

A few investigators have explored alternative propulsion schemes. Lighthill's studies[1] of fish and fish fin propulsion are well known. Bowlus[2] adapted such motions to build a prototype "swimming" boat. Manually operated "fish fin" propulsors are available for small boat propulsion. Muench and Garrett[3] have reviewed some attempts to adapt air breathing pulsejet technology to water propulsion. Dickmann[4] and Siekmann[5] have examined the wake from such devices in some detail.

The present section is concerned with a generalized pulsed propulsion device, which expels a series of pulses to develop forward acting thrust. We can suppose that a fluid oscillates in a tube, as shown in Figure 20 and we may imagine that the oscillation is forced by a piston in some way, which is immaterial to the investigation.

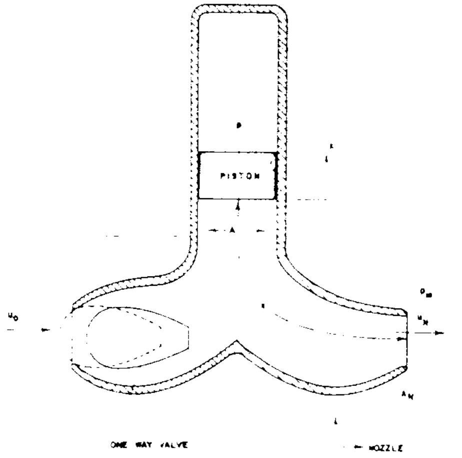


FIG. 20. A GENERALIZED PULSEJET CAPABLE OF RAM RECOVERY, AND EQUIPPED WITH A NOZZLE.

The Continuous Flow Propulsor

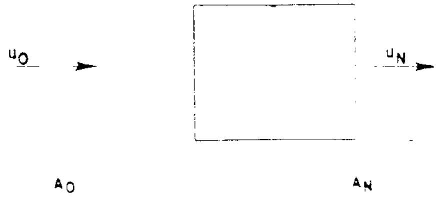


FIG. 21. A ONE-DIMENSIONAL CONTINUOUS FLOW PROPULSOR

In a variation of Froude's classic "actuator disc" analysis, it is easy to derive certain basic relationships for the continuous flow propulsor of Figure 21, working in an incompressible fluid. The thrust is

---

* Fulton's engine was designed and built by James Watt, and it was the third engine exported by Messrs. Boulton, Watt & Company [illegible] A. Baker, The Trockare, "The Engine Powered Vessel"). This was in 18

$$T = \dot{m}(u_N - u_o) = \rho A_N u_N(u_N - u_o) \qquad (1)$$

The additional kinetic energy imparted to the jet is

$$E = \frac{1}{2}\dot{m}(u_N^2 - u_o^2) = \frac{1}{2}\rho A_N u_N(u_N^2 - u_o^2) \qquad (2)$$

If this energy is supplied at no loss, the propulsive efficiency is

$$\eta_p = \frac{Tu_o}{E} = \frac{2u_o}{u_o + u_N} = \frac{2}{1 + \lambda} \qquad (3)$$

where $\lambda = u_N/u_o$.

Thus $\eta_p \to 1.0$ as $T \to 0$.

We can define a thrust coefficient

$$C_T = \frac{T}{\rho u_o^2 A_N} = \frac{u_N}{u_o}\left(\frac{u_N}{u_o} - 1\right) = \lambda(\lambda - 1) \qquad (4)$$

so that

$$\lambda = \frac{1 + \sqrt{1 + 4C_T}}{2}$$

It then follows from equation (3) that

$$\eta_p = \frac{2}{1 + \lambda} = \frac{4}{3 + \sqrt{1 + 4C_T}} \qquad (5)$$

Typical values for C_T range downwards from infinity (when the vessel is stationary) to $C_T = 0.33$ for a destroyer at 26 knots. C_T for a typical waterjet would be an order of magnitude greater at the same speed.

The Pulsed Propulsor

The pulsejet of Figure 20 discharges a finite mass of fluid per cycle. The instantaneous rate of discharge is

$$\frac{dm}{dt} = \rho A_N u_N \qquad (6)$$

$$\therefore m = \rho A_N \int_{\text{discharge}} u_N \, dt \qquad (7)$$

and the exhaust momentum (or impulse I) is

$$I = mv = \rho A_N \int u_N^2 \, dt \qquad (8)$$

When the unit is moving forward at a speed u_o, the total impulse is

$$I = \rho A_N \int_{\text{discharge}} (u_N^2 - u_N u_o)\, dt \qquad (9)$$

The additional kinetic energy imparted to the discharge is

$$\frac{dE_I}{dt} = \frac{1}{2}\rho A_N u_N(u_N^2 - u_o^2)$$

$$\therefore E_I = \frac{1}{2}\rho A_N \int_{\text{discharge}} u_N(u_N^2 - u_o^2)\, dt \qquad (10)$$

If P is the period of one cycle, propulsive efficiency can be written as

$$\eta_p = \frac{I}{P}u_o\frac{P}{E_I} = \frac{I}{E_I}u_o = \frac{2u_o\int(u_N^2 - u_N u_o)\,dt}{\int u_N(u_N^2 - u_o^2)\,dt} \qquad (11)$$

or if $\lambda = u_N/u_o$

$$\eta_p = \frac{2\int(\lambda^2 - \lambda)\,dt}{\int(\lambda^3 - \lambda)\,dt} \qquad (12)$$

where $\lambda = f(t)$ must be zero at the beginning and end of the stroke. If λ = constant, then

$$\eta_p = \frac{2\lambda(\lambda - 1)}{\lambda(\lambda^2 - 1)} = \frac{2}{1 + \lambda} \qquad (13)$$

a result which is identical with equation (3) for a continuous flow propulsor. Before examining other forms of discharge, let us transform equation (12) by writing $d\tau = dt/\phi P$, where the total discharge time is ϕP. Thus the exhaust cycle will be from $\tau = 0$ to $\tau = 1$. Then

$$\eta_p = \frac{2\int_0^1(\lambda^2 - \lambda)\,d\tau}{\int_0^1(\lambda^3 - \lambda)\,d\tau} \qquad (14)$$

Similarly, the thrust coefficient is

$$C_T = \frac{T_{AV}}{\rho u_o^2 A_N \phi} = \int_0^1(\lambda^2 - \lambda)\,d\tau \qquad (15)$$

These integrations have been performed for some simple exhaust velocity profiles, with results tabulated in Table 2 and plotted in Figure 22. Note that when the velocity variation $\lambda' = f(\tau)$ is superimposed on a constant value $\lambda = 1$, $\eta_p \to 1$ as $\Gamma \to 1$. That is, as the function tends to a uniform exhaust velocity ($u_N = u_o$.) In the more practical case, when $\lambda(\tau)$ rises from and falls to zero, there is a definite maximum in η_p at a finite C_T. The more uniform the velocity profile, the higher is this maximum.

In comparison with a steady flow waterpump (C_T = 1 to 5 say) it is clear that the pulsejet does not experience a severe performance penalty. In many practical pulsejet applications, the velocity variation (a) of Figure 22 is achievable, so that at $C_T = 2$, $\eta_p = 0.6$, vis-a-vis $\eta_p = 0.67$ for the continuous flow pump. This is not a major factor in the total trade-off when other considerations, such as cost and mechanical simplicity and real viscous flow losses are of importance.

Internal Flow Equations of a Pulsed Flow Engine

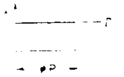
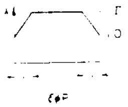
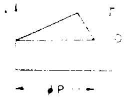
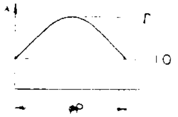
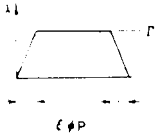
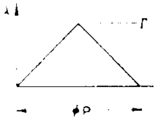
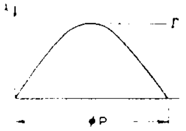
The preceding analysis applies to any pulsejet, liquid or air-breathing, because it is based only on what happens, external to the engine. In what follows, we consider the internal flow, but restrict the analysis to the incompressible case only. For generality in the water case, we consider the liquid to be heavy. The interface is assumed to be the same distance x from both inlet and exhaust.

Regardless of the mechanism used to vary the pressure (p) above the piston, in the propulsor of Figure 20, the fluid column in the duct will be the "linear flywheel" with which it reacts. The pressure (p) will influence the motion of the fluid and vice versa.

For simplicity, let us imagine that the fluid piston interface is a distance x from both the inlet plane and the exit plane, that the piston is weightless [illegible]

Table 2. Idealized Exhaust Velocity - Time Histories Where $u_N \geq u_o$

$\lambda = u_N/u_o$, $\Gamma = \lambda_{max}$, η = Propulsive Efficiency = $\frac{Tu_o}{\text{K.E. In Jet}}$, $C_T = T/\rho u_o^2 A_n\phi$, Γ_{opt} = value for maximum η

UNIFORM VELOCITY 	$C_T = \Gamma(\Gamma - 1)$	$\eta = \frac{2}{1+\Gamma}$	$\Gamma_{OPT} = 1$, $\eta_{OPT} = 1$, $C_{T_{OPT}} = 0$
TRAPEZOIDAL SUPER-VELOCITY 	$C_T = (\Gamma - 1)\left[(1 - \frac{4}{3}\zeta)\Gamma + \frac{1}{3}\zeta\right]$	$\eta = \frac{2\left[(1 - \frac{4}{3}\zeta)\Gamma + \frac{1}{3}\zeta\right]}{(\Gamma + 1)\left[(1 - \frac{2}{3}\zeta)\Gamma + \frac{1}{2}\zeta\right]}$	$\Gamma_{OPT} = 1$, $\eta_{OPT} = 1$, $C_{T_{OPT}} = 0$
TRIANGULAR SUPER-VELOCITY 	$C_T = (\Gamma - 1)(\frac{1}{3}\Gamma + \frac{2}{3})$	$\eta = \frac{(\frac{2}{3}\Gamma + \frac{1}{3})}{\frac{1}{4}(\Gamma + 1)^2}$	$\Gamma_{OPT} = 1$, $\eta_{OPT} = 1$, $C_{T_{OPT}} = 0$
SINE PULSE SUPER-VELOCITY 	$C_T = \frac{1}{2}(\Gamma - 1)\left[(\Gamma - 1) + \frac{4}{\pi}\right]$	$\eta = 2\left[\frac{\frac{\pi}{4}(\Gamma - 1) + 1}{\frac{1}{3}(\Gamma - 1)^2 + \frac{3\pi}{8}(\Gamma - 1) + 1}\right]$	$\Gamma_{OPT} = 1$, $\eta_{OPT} = 1$, $C_{T_{OPT}} = 0$
TRAPEZOIDAL VELOCITY 	$C_T = \Gamma\left[(\Gamma - 1) - (\frac{4}{3}\Gamma - 1)\zeta\right]$	$\eta = 2\left[\frac{(\Gamma - 1) - (\frac{4}{3}\Gamma - 1)\zeta}{(\Gamma^2 - 1) - (\frac{3}{2}\Gamma^2 - 1)\zeta}\right]$	$\Gamma_{OPT} =$ $\eta_{OPT} =$ $C_{T_{OPT}} =$
TRIANGULAR VELOCITY 	$C_T = \frac{1}{2}\Gamma(\frac{2}{3}\Gamma - 1)$	$\eta = \frac{2(\frac{2}{3}\Gamma - 1)}{(\frac{1}{2}\Gamma^2 - 1)}$	$\Gamma_{OPT} = 2$, $\eta_{OPT} = \frac{2}{3}$, $C_{T_{OPT}} = \frac{1}{3}$
SINE PULSE VELOCITY 	$C_T = \frac{\Gamma}{2}(\Gamma - \frac{4}{\pi})$	$\eta = 2\left[\frac{\frac{\pi}{4}\Gamma - 1}{\frac{2}{3}\Gamma^2 - 1}\right]$	$\Gamma_{OPT} = \frac{4}{\pi}\left[1 + \sqrt{1 - \frac{3\pi^2}{32}}\right] = 1.6213$, $\eta_{OPT} = 0.7266$, $C_{T_{OPT}} = 0.282$

p = pressure above the interface or piston
p_1 = inlet static pressure
p_N = static pressure at the nozzle [illegible]
h_i = inlet plane depth below the surface
A_N = nozzle area
A_D = duct area away from nozzle = constant in this analysis
η_R = ram recovery efficiency

In the intake stroke the static pressure at the [illegible] (from Bernoulli):

[illegible]

The mass of fluid in the tube is

[illegible]

Then the equation of inlet motion is

$$A_D(p_i - p) = \frac{d}{dt}(m\dot{x}) \qquad (18)$$

$$= m\ddot{x} + \dot{m}\dot{x} \qquad (19)$$

Now if $\dot{x} < u_o$, the second term will vanish, since the momentum is already in the water. The velocity relative to water axes is $v = \dot{x} - u_o$. Thus we may say

$$A_D(p_i - p) = m\ddot{x} + \dot{m}(\dot{x} - u_o) \qquad \dot{x} > u_o$$

$$A_D(p_i - p) = m\ddot{x} \qquad \dot{x} < u_o$$

Now $m = \rho A_D x$, so that substituting equation (16) leads to

$$\frac{p_o - p}{\rho} + gh_i + \frac{\eta_R}{2}u_o^2 = x\ddot{x} + \frac{1}{2}(\dot{x}^2 - \dot{x}u_o)$$

$$\frac{p_o - p}{\rho} + gh_i + \frac{\eta_R}{2}u_o^2 = x\ddot{x} + \frac{1}{2}\dot{x}^2$$

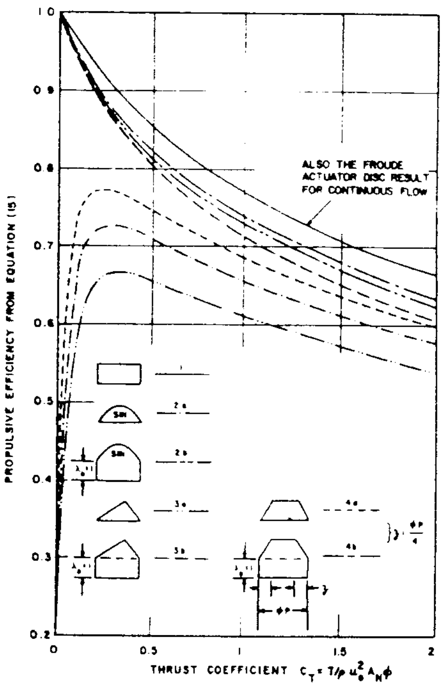


FIG. 22. IDEAL PROPULSIVE EFFICIENCY OF A PULSEJET, AS A FUNCTION OF ITS EXHAUST VELOCITY TIME HISTORY SHAPE. E.G. FOR PULSE SHAPE 2A,
$\lambda = U_{EXHAUST}/U_O =$ [illegible]SIN[(π/ϕ)(T/P)], WHERE
ϕ = DISCHARGE TIME/TOTAL PERIOD (P)
T = TIME FROM START OF DISCHARGE

(Note that the assumption

$$A_o(p_\infty - p) = \frac{d}{dt}(m\dot{x})$$

leads to

$$\frac{p_\infty + p}{\rho} + gh_i + \frac{n_R}{2}u_o^2 = x\ddot{x} + (\dot{x})^2$$

- the factor 1/2 being absent on the right hand side. This mistake was made in a number of our earlier investigations.)

For the exhaust stroke the back-pressure outside the nozzle is assumed to be $p_\infty - \rho g h_N$. This is conservative when $|\dot{x}| < u_o$, since a lower pressure would then be experienced, due to wake effects.

During a discharge, the fluid flows through a nozzle of local area A and terminal area A_N. The velocity at any point x in the nozzle is (using x_i to denote the interface location in the next three lines only):

$$\dot{x} = \dot{x}_i A_o/A \qquad \text{(by continuity)}$$

$$\therefore \quad \frac{d(m\dot{x})}{dx} = \rho A\dot{x} = \rho A_o \dot{x}_i$$

$$\therefore \quad m\dot{x} = \rho A_o \dot{x}_i \int dx = \rho A_o x_i \dot{x}_i \tag{22}$$

- a result which is identical with (18); i.e. $m\dot{x} = (\rho A_o x)\dot{x}$ in the notation of equation (18).

From Bernoulli, the static pressure just upstream of the nozzle is given by

$$p_N + \frac{1}{2}(\dot{x})^2 = p_\infty + \rho g h_N + \frac{1}{2}\rho u_N^2$$

$$= p_\infty + \rho g h_N + \frac{1}{2}\rho(\dot{x})^2(A_o/A_N)^2$$

$$p_N = p_\infty + \rho g h_N + \frac{1}{2}\rho(\dot{x})^2\zeta \tag{23}$$

where

$$\zeta = \left[\left(\frac{A_o}{A_N}\right)^2 - 1\right]$$

As before

$$A_o(p_N - p) = \frac{d}{dt}(m\dot{x})$$

In the discharge case

$$\frac{d}{dt}(m\dot{x}) = m\ddot{x}$$

because momentum is conserved in the jet ($\dot{m} = 0$). Thus

$$\frac{p_\infty - p}{\rho} + gh_N = x\ddot{x} - \frac{1}{2}\zeta(\dot{x})^2 \tag{24}$$

When water pulsejet engines are operated on surface craft, h will be of the order of a foot or two, and hence this term may generally be neglected on both intake and exhaust strokes. It may always be neglected for air-breathing pulsejets.

Can the Condition $\dot{x} > u_o$ Exist During Induction

Equation (21) ($\dot{x} < u_o$) may be written as

$$P = x\ddot{x} + \frac{1}{2}(\dot{x})^2 \tag{25}$$

The transformation $x = z^{2/3}$ transforms this equation to

$$P = \frac{2}{3}z^{1/3}\ddot{z} \tag{26}$$

and after some manipulation

$$\dot{x} = \sqrt{2P(1 - x_o/x)} \tag{27}$$

x_o being the initial value at the start of the induction stroke. Substituting for P and neglecting h, we see that if $p = p_\infty$

$$\dot{x} = u_o\sqrt{n_R(1 - x_o/x)} \tag{28}$$

so that $\dot{x} < u_o$, always. If, on the other hand, p = 0 (vacuum)

$$\dot{x} = \sqrt{\left(\frac{2p_\infty}{\rho} + n_R u_o^2\right)\left(1 - \frac{x_o}{x}\right)}$$

$$= u_o\sqrt{\left(\frac{p_\infty}{\frac{1}{2}\rho u_o^2} + n_R\right)\left(1 - \frac{x_o}{x}\right)} \tag{29}$$

so that, in some cases $\dot{x} > u_o$. (In water, p_∞ corresponds to u_o = 46 ft/sec dynamic pressure at sea level.)

Pressure and Viscous Losses

The loss due to bends, etc. is expressed in the form

$$\Delta p_B = \pm\phi\frac{1}{2}\rho(\dot{x})^2 \tag{30}$$

- and the sign must be reversed for each direction of motion. A crude skin friction loss estimate can be obtained from the Darcy-Weisbach equation

$$h_F = \frac{\Delta p_{SF}}{\rho g} = \pm f\left(\frac{x}{D}\right)\frac{(\dot{x})^2}{2g}$$

whence

$$\Delta p_{SF} = \pm f\left(\frac{x}{D}\right)\frac{1}{2}\rho(\dot{x})^2 \tag{31}$$

or in terms of a skin friction coefficient C_f

$$\Delta p_{SF} = \frac{\pm C_f \times \pi D \frac{1}{2}\rho(\dot{x})^2}{\frac{\pi D^2}{4}}$$

$$= \pm 4C_f(x/D)\frac{1}{2}\rho(\dot{x})^2 \tag{32}$$

Thus $C_f = f/4$. For smooth pipes, and $R_e = \dot{x}D/\nu$, $R_e > 1000$

$$f = \frac{1}{[1.778\log_{10}R_e - 1.383]^2} \tag{33}$$

For R < 1000,

$$f = \frac{64}{R_e} \tag{34}$$

and therefore, from equation (31)

$$\Delta p_{SF} = \frac{64\nu}{\dot{x}D}(x/D)\frac{1}{2}\rho(\dot{x})^2$$

$$= 32\nu\rho\dot{x}\frac{x}{D^2} = 32\mu\frac{\dot{x}x}{D^2} \tag{35}$$

Equations (33) and (34) are plotted in Figure 23. These equations assume, of course, that steady flow results are applicable in an oscillating column; an assumption which has yet to be tested.

Summary of Equations, Including Internal Losses

Instroke -- $R_e > 1000$, $\dot{x} < u_o$

$$\frac{p_\infty - p}{\rho} + gh_i + \frac{n_R}{2}u_o^2 - \phi\frac{1}{2}(\dot{x})^2 - f\frac{x}{D}\frac{1}{2}(\dot{x})^2 = x\ddot{x} + \frac{1}{2}(\dot{x})^2$$

or

$$\frac{p_\infty - p}{\rho} + gh_i + \frac{n_R}{2}u_o^2 = x\ddot{x} + \frac{1}{2}(\dot{x})^2[1 + f(x/D) + \phi] \tag{36}$$

Instroke -- $R_e < 1000$, $\dot{x} < u_o$

$$\frac{p_\infty - p}{\rho} + gh_i + \frac{n_R}{2}u_o^2 - \phi\frac{1}{2}(\dot{x})^2 - 32\nu\rho\frac{x}{D^2}\dot{x} = x\ddot{x} + \frac{1}{2}(\dot{x})^2$$

or

$$\frac{p_\infty - p}{\rho} + gh_i + \frac{n_R}{2}u_o^2 = x\ddot{x} + \frac{1}{2}(\dot{x})^2[1 + \phi] + \frac{32\nu}{D^2}x\dot{x} \tag{37}$$

Outstroke -- $R_e > 1000$

$$\frac{p_\infty - p}{\rho} + gh_N + \phi\frac{1}{2}(\dot{x})^2 + f\frac{x}{D}\frac{1}{2}(\dot{x})^2 = x\ddot{x} - \frac{1}{2}\zeta(\dot{x})^2$$

or

$$\frac{p_\infty - p}{\rho} + gh_N = x\ddot{x} - \frac{1}{2}(\dot{x})^2[\zeta + fx/D + \phi] \tag{38}$$

TURBULENT

LAMINAR

$1/\sqrt{f} = \sqrt{R}/8$

$1/\sqrt{f} = 1.778\log_{10}(R) - 1.383$

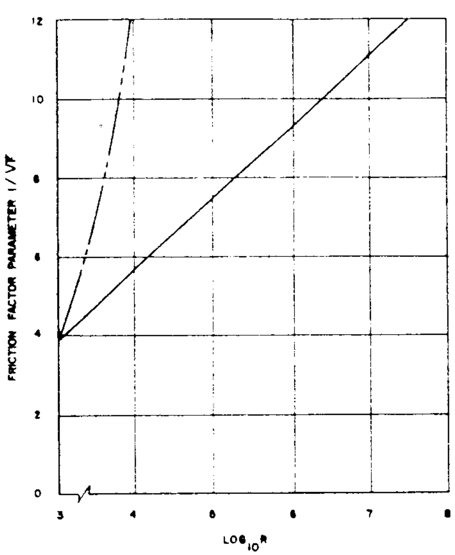


FIG. 23. LAMINAR AND FULLY TURBULENT FRICTION FACTORS FOR A SMOOTH PIPE. R = uD/ν, THE PIPE REYNOLDS NUMBER.

where

$$\zeta = (A_o/A_N)^2 - 1$$

Outstroke -- $R_e < 1000$

$$\frac{p_\infty - p}{\rho} + gh_N + \phi\frac{1}{2}(\dot{x})^2 - \frac{32\nu}{D^2}\rho x\dot{x} = x\ddot{x} - \frac{1}{2}\zeta(\dot{x})^2$$

or

$$\frac{p_\infty - p}{\rho} + gh_N = x\ddot{x} - \frac{1}{2}(\dot{x})^2[\zeta + \phi] + \frac{32\nu}{D^2}\rho x\dot{x} \tag{39}$$

where

$$\zeta = (A_o/A_N)^2 - 1$$

Solutions of the Equations

Even if the pressure and viscous loss terms are ignored, the equations are still very nonlinear, i.e.

For the instroke

$$\frac{P - p}{\rho} = x\ddot{x} + \frac{1}{2}(\dot{x})^2 \tag{40}$$

For the outstroke

$$\frac{P - p}{\rho} = x\ddot{x} - \frac{1}{2}\zeta(\dot{x})^2 \tag{41}$$

If the piston motion is defined, as when driven by a crank, then solutions are easy to obtain. The case of sinusoidal motion is worked in the Appendix, as an example.

When the forcing pressure is defined, the equations must be solved for x. Fortunately they can be linearized. For the instroke equation the substitution

$$x = z^{2/3} \qquad (42)$$

transforms equation (40) to

$$\ddot{z} = \frac{3}{2} z^{-1/3} \frac{P - p}{\rho} \qquad (43)$$

The first integration (between $\dot{z}_1$ and $\dot{z}_2$) gives

$$\frac{1}{3}(\dot{z}_2^{\,2} - \dot{z}_1^{\,2}) = \frac{3}{2}\frac{P}{\rho}(z_2^{\,3/2} - z_1^{\,3/2}) - \frac{1}{\rho}\int_{z_1}^{z_2} \frac{p}{z^{1/3}}\, dz$$

Transferring back to the x domain

$$\frac{1}{2}\rho(\dot{x}_2^{\,2}x_2 - \dot{x}_1^{\,2}x_1) = P(x_2 - x_1) - \int_{x_1}^{x_2} p\, dx \qquad (44)$$

Since the instroke generally starts with $\dot{x}_1 = 0$

$$\frac{1}{2}\rho\dot{x}_2^{\,2} = P\left(1 - \frac{x_1}{x_2}\right) - \frac{1}{x_2}\int_{x_1}^{x_2} p\, dx \qquad (45)$$

For many cases

$$p = p_1\left(\frac{x - x_1}{x - x_2}\right)^n \qquad (46)$$

where

$n = \gamma$ for adiabatic compression

$= 1$ for isothermal

$= 0$ when the cylinder is vented to ambient

All these cases are simply integratable.

Equation (41), the outstroke equation, is linearized by the substitution

$$x = z^{2/(2-\zeta)} \qquad (47)$$

which leads to

$$\ddot{z} = \left(\frac{2-\zeta}{2}\right) z^{(\zeta+2)/(\zeta-2)} (P - p/\rho) \qquad (48)$$

The first integration in the z domain gives

$$\left(\frac{1}{2-\zeta}\right)\left(\dot{z}_2^{\,2} - \dot{z}_1^{\,2}\right) = \frac{P}{\rho}\left(\frac{\zeta-2}{2\zeta}\right)\left(z_2^{\,2\zeta/(\zeta-2)} - z_1^{\,2\zeta/(\zeta-1)}\right) - \frac{1}{\rho}\int_{z_1}^{z_2} z^{(\zeta-2)/(\zeta+2)}\, dz$$

Transferring to the x domain

$$\frac{1}{2}\rho(\dot{x}_2^{\,2}x_2^{\,-\zeta} - \dot{x}_1^{\,2}x_1^{\,-\zeta}) = -\frac{P}{\zeta}(x_2^{\,-\zeta} - x_1^{\,-\zeta}) - \int_{x_1}^{x_2} p x^{-(\zeta+1)}\, dx \qquad (49)$$

When $\dot{x}_1 = 0$ at the start of the exhaust stroke, this simplifies to

$$\frac{1}{2}\rho\dot{x}_2^{\,2} = \frac{P}{\zeta}\left[1 - \left(\frac{x_1}{x_2}\right)^{-\zeta}\right] - x_2^{\,\zeta}\int_{x_1}^{x_2} p x^{-(\zeta+1)}\, dx \qquad (50)$$

The pressure variations of equation (46) can be integrated for n = 1, but for n = γ it is necessary to substitute a series for the function $px^{-(\zeta+1)}$ in order to integrate.

In many practical cases, of course, it is more convenient to solve the complete equations numerically.

Appendix

Viscous Losses in a Sinusoidally-Driven Pulsejet at Zero Forward Speed

$$\left.\begin{aligned} &\text{Let } x = X_M + X_o \sin \omega t \\ &\text{so that } \dot{x} = X_o \omega \cos \omega t \\ &\text{and } \ddot{x} = -X_o \omega^2 \sin \omega t \end{aligned}\right\} \qquad (A\text{-}1)$$

Substituting in equation (36) for the instroke

$$\begin{aligned}\frac{p_\infty - p}{\rho} = &- X_M X_o \omega^2 \sin \omega t - X_o^{\,2}\omega^2 \sin^2 \omega t \\ &+ \frac{1}{2} X_o^{\,2}\omega^2 \cos^2 \omega t(1 + \phi_{IN}) + \frac{1}{2}\frac{X_o^{\,2}}{D} X_M f\omega^2 \cos^2 \omega t \\ &+ \frac{1}{2}\frac{X_o^{\,3}}{D} f\omega^2 \cos^2 \omega t \sin \omega t\end{aligned}$$

The average thrust for the instroke is

$$\begin{aligned}-\frac{T_{IN}}{\rho} &= \frac{1}{2\pi}\int_{-\pi/2}^{\pi/2} \frac{p_\infty - p}{\rho}\, d(\omega t) \\ &= \frac{1}{2\pi}\Bigg[X_M X_o \omega^2 \cos \omega t - \frac{X_o^{\,2}\omega^2}{2}(\omega t - \sin \omega t \cos \omega t) \\ &\quad + \frac{1}{4} X_o^{\,2}\omega^2(1 + \phi_{IN})(\omega t + \sin \omega t \cos \omega t) \\ &\quad + \frac{1}{4}\frac{X_o^{\,2}X_M}{D} f\omega^2(\omega t + \sin \omega t \cos \omega t) \\ &\quad - \frac{1}{6}\frac{X_o^{\,3}}{D} f\omega^2 \cos^3 \omega t\Bigg]_{-\pi/2}^{\pi/2} \\ &= \frac{1}{2\pi}\left[-\frac{\pi}{2}X_o^{\,2}\omega^2 + \frac{\pi}{4}X_o^{\,2}\omega^2(1+\phi) + \frac{\pi}{4}\frac{X_o^{\,2}X_M}{D} f\omega^2\right] \\ &= \frac{X_o^{\,2}\omega^2}{8}\left[-1 + \phi_{IN} + \frac{X_M f}{D}\right]\end{aligned}$$

$$T_{IN} = \frac{1}{8}\rho X_o^{\,2}\omega^2\left(1 - \phi_{IN} - \frac{X_M}{D} f\right) \qquad (A\text{-}2)$$

Substituting in equation (38) for the outstroke

$$\begin{aligned}\frac{p_\infty - p}{\rho} = &- X_M X_o \omega^2 \sin \omega t - X_o^{\,2}\omega^2 \sin^2 \omega t \\ &- \frac{1}{2} X_o^{\,2}\omega^2 \cos^2 \omega t(\zeta + \phi_{OUT}) - \frac{1}{2} X_o^{\,2}\frac{X_M}{D} f\omega^2 \cos^2 \omega t \\ &- \frac{1}{2}\frac{X_o^{\,3}}{D} f\omega^2 \cos^2 \omega t \sin \omega t\end{aligned}$$

$$\begin{aligned}-\frac{T_{OUT}}{\rho} &= \frac{1}{2\pi}\int_{\pi/2}^{3\pi/2} \frac{p_\infty - p}{\rho}\, d(\omega t) \\ &= \frac{1}{2\pi}\left[-\frac{\pi}{2}X_o^{\,2}\omega^2 - \frac{\pi}{4}X_o^{\,2}\omega^2(\zeta + \phi_{OUT}) - \frac{\pi}{4}X_o^{\,2}\frac{X_M}{D} f\omega^2\right]\end{aligned}$$

$$T_{OUT} = \frac{1}{8}\rho X_o^{\,2}\omega^2\left(2 + \zeta + \phi_{OUT} + \frac{X_M}{D} f\right) \qquad (A\text{-}3)$$

$$T_{AV} = T_{IN} + T_{OUT} = \frac{1}{8}\rho X_o^{\,2}\omega^2(3 + \zeta + \phi_{OUT} - \phi_{IN}) \qquad (A\text{-}4)$$

For the case of a parallel duct ($\zeta = 0$, $\phi_{OUT} = 0$), the contraction loss at the inlet is $\phi_{IN} = 1/2$, so that the equations give

$$\left.\begin{aligned} T_{IN} &= \frac{1}{8}\rho X_o^{\,2}\omega^2\left(\frac{1}{2} - \frac{X_M}{D} f\right) \\ T_{OUT} &= \frac{1}{8}\rho X_o^{\,2}\omega^2\left(\frac{3}{2} + \frac{X_M}{D} f\right) \\ T_{AV} &= \frac{1}{4}\rho X_o^{\,2}\omega^2 \end{aligned}\right\} \qquad (A\text{-}5)$$

Note that the skin friction terms do not influence the average thrust, so long as the friction factor f can be regarded as a constant. Equation A-5 is compared with some experimental measurements of average thrust in Figure A-1.

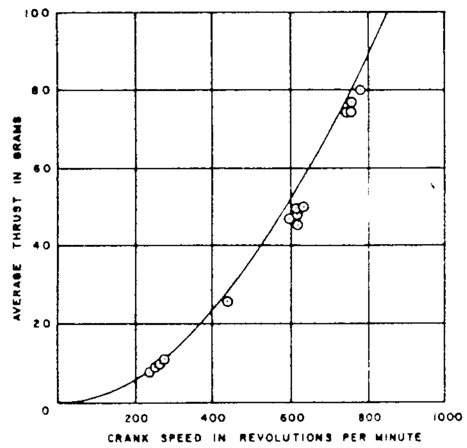


FIG. A-1. EQUATION (A-5) COMPARED WITH THE EXPERIMENTAL MEASUREMENTS OF TRUNZ, REPORTED BY DICKMANN.[4]

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Schiffsantrieb mit instationären Vortriebsorganen

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1. Einleitung.

Die Entwicklung der Vortriebsorgane im Flugzeugbau hat bekanntlich zu einer weitgehenden Verdrängung der Luftschrauben zugunsten von Strahltriebwerken geführt. Dies legt eine Überprüfung nahe, ob auch bei Schiffen der erprobte Schraubenpropeller — wenigstens in Sonderfällen — durch andersartige Organe ersetzt werden kann; wenn auch die den Strahltriebwerken eigene, direkte Reaktionswirkung der Verbrennungsgase für die Anwendung auf Schiffen, die in dem 800mal schwereren Wasser schwimmen, von vornherein ausscheiden dürfte, so sollte eine solche Untersuchung doch möglichst auch ein Verfahren einschließen, bei dem der Dampf oder die Verbrennungsgase direkt auf diejenige Wassermenge einwirken, deren Impuls den zum Vortrieb notwendigen Schub ergibt.

Es besteht wohl kein Zweifel darüber, daß der hochgezüchtete Schraubenpropeller bei sonst gleichen Bedingungen (gleiche Strahlfläche) von neuartigen Organen hinsichtlich des Propulsionswirkungsgrades kaum übertroffen werden dürfte. Immerhin sind auch noch andere nützliche Ziele denkbar, wie z. B.: Vereinfachung der Maschinenanlage; günstige räumliche Unterbringung der Stralfläche unter möglichster Ausnutzung des vollen Hauptspantquerschnittes und damit doch eine Vergrößerung des Propulsionswirkungsgrades; Vergrößerung der (bisher durch Kavitation begrenzten) Flächenbelastung und anderes.

Die nachstehende Studie greift zwei Möglichkeiten der Vortriebserzeugung auf, von denen wohl wenigstens die zweite bisher noch kaum erörtert worden ist.

Unter bewußtem Verzicht auf eine an sich mögliche genauere hydrodynamische Rechnung durch Lösung der auftretenden Integralgleichungen wird versucht, die Vorgänge in ihrem Bewegungsablauf anschaulich zu machen und dann hieraus mit einfachen Ansätzen die Größenordnung der Kraftwirkungen abzuschätzen.

Erforderliche Hilfsrechnungen sind in einem Anhang zusammengefaßt.

2. Der nichtstationäre Idealpropeller.

Die in Ziffer 3) und 4) zu untersuchenden Propeller arbeiten mit wechselnder, d. h. nicht stationärer Kraftwirkung. Eine genaue Betrachtung solcher Vorgänge bietet bekanntlich zusätzliche rechnerische Schwierigkeiten, da in jedem Zeitpunkt t und an jedem Ort (x, y, z) des ausgedehnten Flüssigkeitsraumes auch die „lokale" Beschleunigung dw/dt des Geschwindigkeitsvektors w (x, y, z, t) im Kräftegleichgewicht an jedem einzelnen Flüssigkeitsteilchen berücksichtigt werden muß.

Handelt es sich jedoch um periodisch wechselnde Vorgänge, die für alle physikalischen Größen, wenigstens für einen gegenüber der Schwankungszeit langen Zeitraum gemessen, gewisse zeitliche Mittelwerte aufweisen — und um solche Vorgänge wird es sich handeln — so gelingt eine gute Vorstellung des Bewegungsablaufes der Strömung und eine abschätzende Ermittlung der entstehenden äußeren Kräfte (Propellerschub) durch recht einfache Überlegungen. Bevor diese Methode auf die Probleme zu 3) und 4) angewendet wird, soll sie an einem einfachen, idealisierenden Beispiel erläutert und bewiesen werden.

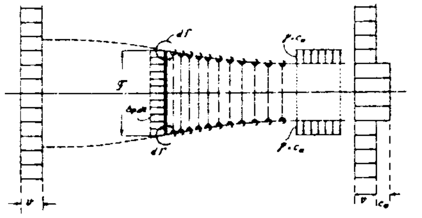


Abb. 1. Der stationäre und der nicht stationäre Idealpropeller. Wirbelentstehung. Strahlquerschnitt rechts = F_a.

Bild 1 zeigt die bekannte Anordnung eines einfachen Strahlpropellers, bei dem in einer Transportströmung v am Ort der Propellerfläche F (durch äußere dort wirkende Kräfte) ein Drucksprung Δp erzeugt wird. Die Ergebnisse sind bekannt. [1] und [2].

Dieses selbe Bild soll jetzt aber anders betrachtet werden. Der Drucksprung wirke nicht kontinuierlich, sondern nur während einer sehr (beliebig) kurzen Zeit dt. In der den Zusammenhang zwischen Flüssigkeitsdruck, Geschwindigkeiten und äußeren Kräften vermittelnden Eulerschen Vektorgleichung können dann die „konvektiven" Beschleunigungsglieder (durch Geschwindigkeitsänderung bei Verschiebung der Teilchen) gegenüber der „lokalen" Beschleunigung (durch Geschwindigkeitsänderung am selben Ort) beliebig klein gemacht und damit vernachlässigt werden. Die Eulergleichung läßt sich dann leicht auflösen. Durch Eliminierung des Flüssigkeitsdruckes hat Burgers [3] folgendes einfache Ergebnis erhalten:

Es entsteht ein einzelner Wirbelring mit der Zirkulation dΓ (in m²/sek) (z. B. Rauchring bei plötzlichem Ausstoßen von rauchiger Luft), dessen Achse mit der Berandung der Fläche F, auf welcher der die sehr kurze Zeit dt dauernde, räumlich aber konstante Drucksprung Δp gewirkt hat, zusammenfällt.

Die Zirkulation oder Wirbelstärke errechnet sich zu

$$d\Gamma = \frac{1}{\varrho}(\Delta p \cdot dt) \quad \ldots \ldots (1)$$

ist also dem Druckimpuls ($\Delta p \cdot dt$) (in kgsec) proportional. ϱ ist die Massendichte. In Bild 1 ist am Rand der Propellerfläche F solch ein Einzelwirbel mit seiner Drehrichtung zeichnerisch angedeutet.

Folgt nun diesem einmaligen Kraftimpuls kein weiterer, so wird der erzeugte Einzelwirbel mit der ihm eigenen (durch die Transportgeschwindigkeit, durch seine Gestalt, seine Zirkulation und durch die Größe des hier nicht näher zu erörternden „echten" Kernwirbels bestimmten) Geschwindigkeit in Richtung der früheren Kraftwirkung (nach rechts) davonschwimmen.

Man kann aber auch umgekehrt — und das ist jetzt besonders wesentlich — aus dem Vorhandensein eines solchen Wirbels darauf schließen, daß irgendwann einmal früher ein Kraftimpuls gewirkt haben muß, ja man könnte sogar durch eine Aufmessung von Gestalt und Zirkulation des Wirbels (sofern er durch andere Einflüsse seine Form nicht verändert hat) nach Gl. 1) auf die genaue Größe des Druckimpulses und auf die Fläche, wo er gewirkt hat, und damit schließlich auf die Größe des Kraftimpulses $P \cdot dt = F \cdot \Delta p \cdot dt$ sowie aus der Drehrichtung und Bewegungsrichtung endlich auch auf die Kraftrichtung schließen. (Die Eigengeschwindigkeit des Wirbels wird — ihrem absoluten Wert nach — nicht benötigt, die Kenntnis eines Kernradius ist also mit Recht nebensächlich).

Vorstehende Überlegung bildet in etwas abgewandelter Form das Kernstück der erwähnten, nunmehr mehrfach anzuwendenden Methode.

Läßt man jetzt viele (immer noch ruckartige) Kraftimpulse einander folgen, so gilt immer noch Gl. 1, die sich auch schreiben läßt:

$$\varrho \frac{d\Gamma}{dt} = \Delta p = f(t) \quad \ldots \ldots (1a)$$

wobei beide Seiten eine (identische) Funktion der Zeit darstellen („Zerhacker-Kurve"). Der Ausdruck $d\Gamma/dt$ läßt sich auffassen als ein „Wirbelfluß". Auf der Propellerberandung werden nämlich jetzt ständig in sich geschlossene Wirbel mit $d\Gamma/dt$ als Wirbelstärke in der Zeiteinheit erzeugt, die in Kraftrichtung abfließen mit irgend einer zunächst unwesentlichen Geschwindigkeit.

Nimmt man nun weiterhin an, daß die — sonst beliebigen — kurzen Druckimpulse doch wenigstens periodisch verlaufen und eine Grundfrequenz $\omega = \frac{2\pi}{T}$ mit der (längsten) Periode $T = \frac{2\pi}{\omega}$ besitzen, so kann man definieren

$$\left.\begin{aligned} \Delta p_m &= \frac{1}{T} \cdot \int_0^T \Delta p \cdot dt \quad \ldots \ldots \\ \text{und } \left(\frac{d\Gamma}{dt}\right)_m &= \frac{1}{T}\int_0^T \frac{d\Gamma}{dt}\, dt = \frac{1}{T}\int_0^T d\Gamma = \frac{\Gamma_t}{T} \quad \ldots \ldots \end{aligned}\right\} (2)$$

und dadurch zeitliche Mittelwerte des Drucksprungs und des Wirbelflusses einführen. Auch dann noch gilt die Gleichung (1a)

$$\varrho \cdot \left(\frac{d\Gamma}{dt}\right)_m = \Delta p_m \quad \ldots \ldots (3)$$

jetzt aber unabhängig von der Zeit (also quasi stationär).

$$\frac{T}{\varrho} \cdot \Delta p_m = \left(\frac{d\Gamma}{dt_m}\right) \cdot T = \Gamma_t \quad \ldots \ldots (2a)$$

ist dabei die während der Zeit T erzeugte Gesamtwirbelstärke.

Läßt man nun die (periodischen) Kraftimpulse wirken, stellt man sich ferner auf einen Beobachtungsposten weit hinter dem Ort der Kraft = Schuberzeugung, wobei man aber in demselben relativen Bezugssystem bleibt, wie der (instationär wirkende) Propeller, d. h. also bei Fahrt (Transportgeschwindigkeit v) ihm mit v nachfährt, so werden die im (runden) Propeller erzeugten Wirbel (-ringe) bei Annahme von Reibungsfreiheit nach einiger Zeit brav dahergeschwommen kommen, und zwar angenähert ähnlich gruppiert, wie sie je nach der Zeitcharakteristik von Δp entstanden sind, in Form von Einzelwirbeln, Wirbelgruppen, verteilten Ringwirbelschichten u. ä. (siehe Bild 1). Wegen der vorausgesetzten Periodizität der äußeren Druckwirkung wird aber die im zeitlichen Mittel bei der Propellerfläche F entstandene Wirbelstärke $\left(\frac{d\Gamma}{dt}\right)_m$ in genau gleicher Größe (wegen der Helmholtzschen Wirbelsätze) auch eine quer zur Kraftrichtung an unserem Beobachtungsort gelegte Kontrollebene passieren müssen.

Auf welchem Wege dieser Wirbelfluß dorthin gelangt ist, wie sich die sich gegenseitig durch Induktion beeinflussenden Wirbelringe bewegt und verformt haben, ist also gleichgültig. Diese Überlegung gilt daher insoweit auch bei stark belasteten und sogar bei Standpropellern ohne jede Komplikation. Man könnte die Kontrollebenen an jede beliebige Stelle hinter dem Propeller legen und müßte dort überall den gleichen mittleren Wirbelfluß $(d\Gamma/dt)_m$ feststellen. Man geht aber zweckmäßig weit strahlabwärts, weil dort die gegenseitigen Störungsgeschwindigkeiten der Wirbel (und die abgeklungenen Störungen von Schiffskörpern) zu einer gleichmäßigen, nicht mehr kontrahierenden Konstellation der Wirbel geführt haben, die bei Wahl eines geeigneten Bezugssystems, das mit $v + w_m$ (der „Gruppengeschwindigkeit" der Wirbel) strahlabwärts geführt wird, als ein (beinahe) festes, stationäres Bild erscheint. In diesem Bild haben sich die Wirbelringe im Mittel auf einer zylindrischen Fläche aufgereiht.

Wie in Anhang A) abgeleitet wird, sind nun gleichmäßig verteilte, zylindrische Wirbelschichten der Zirkulationsbelegung γ (in m/sek) anderen Anordnungen mit Einzelwirbeln auf den gleichen Zylinderflächen hydrodynamisch, d. h. hinsichtlich ihres Impulses gleichwertig, sofern der Wirbelfluß $(d\Gamma/dt)_m$ bei gleichen Kontrollflächen, d. h. gleichen Bedingungen dieser Ebenen in großer Entfernung von der Achse, dieselbe Größe hat.

Im Beispiel des instationären, aber periodischen Idealpropellers führt also der Wirbelfluß zu einer Kreiswirbelschicht, wobei gilt:

$$\left(\frac{d\Gamma}{dt}\right)_m = \gamma\,(v + w_m) = \gamma\,(v + \gamma/2)$$

Die Wirbelbelegungsstärke γ ist dabei identisch mit der in Abb. 1 benannten Absolutgeschwindigkeit c_a, am Austritt des betrachteten Flüssigkeitsraumes, vgl. Anhang A, Bild 16b, Gleichung 16a); es gilt also

$$\left(\frac{d\Gamma}{dt}\right)_m = c_a\left(v + \frac{c_a}{2}\right) = v^2\left[\frac{c_a}{v}\left(1 + \frac{c_a}{2v}\right)\right] \quad \ldots (4)$$

Folgt man nun der oben angedeuteten Methode und nimmt man an, daß man einen Wirbelfluß nach obiger Gleichung festgestellt hat, so kann man jetzt nach Gl. (3) eindeutig folgern, daß ein Schuberzeugender Drucksprung Δp_m nach Gl. (3) vorhanden sein muß:

$$\Delta p_m = \varrho\left(\frac{d\Gamma}{dt}\right)_m = \varrho c_a\left(v + \frac{c_a}{2}\right)$$

oder ein mittlerer Schub in kg.

$$S_m = \Delta p_m \cdot F = \varrho \cdot F \cdot c_a\left(v + \frac{c_a}{2}\right) \quad \ldots \ldots (5)$$

Es ist bemerkenswert, daß die Querschnittsfläche F_a der Wirbelschicht oder des Strahls (siehe Bild 1), die keineswegs gleich F ist, bisher nicht benötigt wurde. Ihre Größe ist mit der Propellerfläche gekoppelt und bestimmt sich aus einem normalen Impulsansatz, wie er in der Theorie des einfachen Strahlpropellers [2] oder bei der Widerstandsermittlung von Körpern aus dem Nachlauf [4] üblich ist:

$$S_m = \varrho \cdot Q \cdot c_a$$
$$Q = F_s\,(v + c_a) \text{ (Wassermenge im Strahl)}$$
$$S_m = \varrho\, F_s\,(v + c_a)\, c_a \quad \ldots\ldots \quad (6)$$

Gl. (5) mit (6) gleichgesetzt, ergibt:

$$\frac{F_s}{F} = \frac{v + c_a/2}{v + c_a} = \frac{v + c_a/2}{v + c_a} = \frac{1 + \frac{1}{2}\frac{c_a}{v}}{1 + c_a/v} \quad \ldots\ldots \quad (7)$$

Dies ist die vom einfachen Strahlpropeller her bekannte Strahlkontraktion, die von 1 bei kleiner Belastung auf ½ beim Standpropeller abnimmt und damit nur von der Flächenbelastung abhängt.

Sofern S und Δp als zeitlich konstant angenommen werden, hat man hiermit alle bekannten Ergebnisse der normalen einfachen Strahltheorie wiedergefunden.

Darüber hinaus aber gelten die Gleichungen auch für die nach Gl. (2) definierten zeitlichen Mittelwerte eines zwar periodisch, aber sonst beliebig instationär arbeitenden Propellers.

Weiterhin gestattet vorstehende Ableitung, aus einem durch Messung oder Rechnung anderweitig ermittelten Wirbelfluß, der zwar periodisch aber sonst beliebig instationär sein kann, die erzeugende, zeitlich nach Gl. (2) gemittelte äußere Druck-Schubwirkung zu errechnen. Für den stationären Fall des Zähigkeitswiderstandes von Körpern ist dieser Gedanke wohl zuerst durch v. Kármán bei Wirbelstraßen mit quantitativem Erfolg durchgeführt worden [5], während seine Anwendung beim Problem der nichtstationären Schuberzeugung die nun folgenden einfachen Überlegungen gestattet.

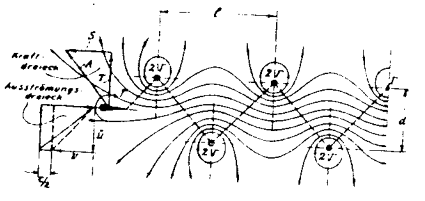


Abb. 2. Prinzip eines Schlagflügelpropellers mit wechselnder, aber konstanter Quergeschwindigkeit. $w = u + \rightarrow (v + c_a/2)$.

3) Der Schlagflügelpropeller

a) Die Flügelbewegung.

Ein Schraubenpropeller arbeitet nach unserer heutigen Anschauungsweise durch den tragflügelartigen Auftrieb seiner einzelnen Blattelemente, wobei jedes dieser Elemente stationär angeströmt wird (sofern keine durch Schiffskörper verursachte Unsymmetrie der Grundströmung vorliegt). Von anderen, nicht stationär beaufschlagten Propellern nach dem Tragflügelprinzip hat sich nur der mechanisch recht komplizierte Voith-Schneider-Propeller durchgesetzt. Offenbar hat die Technik bisher von einer weiteren, ebenso einfachen wie wirkungsvollen Möglichkeit der Vortriebserzeugung keinen Gebrauch gemacht, obwohl diese in der Natur von Vögeln und vor allem von Fischen fast ausschließlich angewendet wird. Dieses Propulsionsorgan soll Schlagflügelpropeller genannt werden.

Wie in Bild 2 gezeigt, bewege sich ein einfacher, hier unendlich lang gedachter Tragflügel mit gerader Skelettlinie genau in deren Richtung mit der gleichmäßigen Fortschrittsgeschwindigkeit v vorwärts. Nun beginne er symmetrisch zu einer solchen mittleren Fortschrittsrichtung zusätzlich in Querrichtung hin und her zu schlagen. Er behalte dabei seine Orientierung im Raume translatorisch bei, führe also keinerlei Drehbewegungen (wie beim Wriggen) aus. Die Schwingungsweite (Doppelamplitude) des Querschlags sei d. Zur Vereinfachung der Überlegung und Rechnung sei hier angenommen, daß die Schläge nach beiden Richtungen jeweils mit konstanter Quergeschwindigkeit u unter plötzlicher Umkehrung in den Hartlagen ausgeführt werden. Der Flügel bewegt sich darin in einem absoluten Bezugssystem auf einer aus geraden Stücken bestehenden Zickzacklinie (gestrichelt in Bild 2) vorwärts. Die Längsabstände l der Umkehrpunkte werden jeweils nach der Periode $T = \frac{l}{v}$ des Schlages erreicht. Diese Periode T bestimmt sich ihrerseits aus der Querbewegung:

$$\frac{T}{2} = \frac{d}{u}, \quad T = \frac{2d}{u} \quad \ldots\ldots \quad (8)$$

b) Das Strömungsfeld.

Das durch die obige einfache Flügelbewegung im absoluten Bezugssystem entstehende Strömungsfeld (im Unendlichen ruhend) läßt sich am besten durch die Induktion der entstehenden Wirbel beschreiben. Der Flügel befindet sich zunächst in Ruhe, und zwar mit seinem scharfen Schwanzende gerade an dem in Abb. 2 oben rechts gezeichneten Ort des natürlich noch nicht vorhandenen ersten Wirbelkerns. Beginnt das Profil nun plötzlich seine Bewegung mit v nach links und u nach unten, so entsteht ein Vorgang, der in Anhang B genau beschrieben ist. Es entsteht eine Anfahrwirbelfläche entlang der Spur des Schwanzendes, die sich vor allem am Anfahrpunkt selbst konzentriert. (vgl. Abb. 20). Wir machen zur Vereinfachung die (die ziemlich gut zutreffende) Annahme, daß sich der Anfahrwirbel genau am Ort des Bewegungsbeginnes in voller Wirbelstärke Γ bildet, die ihrerseits etwas geringer ist als die Zirkulation Γ_s eines mit demselben v und u langzeitig stationär bewegten Flügels.

Das Verhältnis Γ/Γ_s hängt ab vom Verhältnis Flügeltiefe zu Schlagamplitude und von der Zeitcharakteristik der Schlaggeschwindigkeit u.

Um den Flügel herrscht während des ersten Schlages eine unter obigen Annahmen konstante Zirkulation genau der Größe Γ, jedoch mit umgekehrter Drehrichtung wie beim Anfahrwirbel. Nachdem nun der erste Umkehrpunkt erreicht ist, muß das Profil zunächst plötzlich anhalten. Würde es dort endgültig stehenbleiben, so müßte sich die Zirkulation des „festen" Stabwirbels in die eines freien „Stopwirbels" verwandeln. Anfahrwirbel und Stopwirbel würden sich dann durch gegenseitige Induktion nach unten rechts, und zwar etwa normal zur bisherigen Flügelbahn fortbewegen. Diesen Vorgang hat Prandtl besonders anschaulich photographieren lassen [4, Tafel 22]. Der Flügel bleibt aber nicht stehen, sondern bewegt sich sofort wieder mit u aufwärts. Es bildet sich hierdurch ein neuer Anfahrwirbel der Wirbelstärke Γ, welcher gleichen Drehsinn hat, wie der Stopwirbel und sich mit ihm sofort zu einem einzigen freien Wirbel der Stärke 2 Γ zusammenschließt. Um das Profil herrscht während dieses 2. Schlages wieder die Zirkulation Γ, aber diesmal in einer mit dem Anfahrwirbel des 1. Schlages (oben rechts) übereinstimmenden Drehrichtung. Auf einer den ganzen bisherigen Vorgang umschließenden Linie ergibt sich die Gesamtzirkulation aller Stab- und freien Wirbel zu Null, wie es nach Helmholtz sein muß. Bei jedem Schlagwechsel wird nun in der vorbeschriebenen Weise stets ein freier Wirbel der Stärke 2 Γ in den aus Bild 2 ersichtlichen Drehrichtungen zurückbleiben, während um den Flügel stets die Zirkulation Γ gleicher absoluter Größe, aber mit jedem Schlagwechsel kenternder Drehrichtung herrscht.

Hat dieser periodische Schlagvorgang lange genug gewirkt, so ist hinter dem Flügel eine Wirbelschleppe nach Art einer Kármánschen Wirbelstraße (aber umgekehrter Drehrichtung) zurückgeblieben. Die Eigenschaften eines solchen Wirbelgebildes sind im Anhang A 4 näher untersucht.

Die in Bild 2 eingezeichneten Linien sind Absolutstromlinien, vgl. auch im Anhang Bild 17.

Während einer vollen Schlagperiode T nach Gl. 8 hat der Flügel je 2 gegenläufig drehende Stabwirbelpaare der Zirkulation Γ gebildet, die sich auf der Spur der Zickzacklinien im Unendlichen (oder bei endlichen Flügellängen spätestens etwa an den Flügelenden) zu geschlossenen, einläufig drehenden „Wirbelringen" schließen. Gemäß den Anschauungen des Abschn. 2) wird also durch den instationären, aber periodischen Vorgang während der Periode T ein mittlerer Wirbelfluß gemäß Gl. 2 $\left(\frac{d\Gamma}{dt}\right)_m$ erzeugt, der hier $\frac{2\Gamma}{T}$ ist oder mit Gl. 8

$$\left(\frac{d\Gamma}{dt}\right)_m = \frac{\Gamma \cdot u}{d} \quad \ldots\ldots \quad (9)$$

Da gemäß Anhang B die Größe von Γ selbst fast ausschließlich von u und nicht von v abhängt, ist also eine bemerkenswerte Feststellung, daß auch der Wirbelfluß nach (9) im wesentlichen nur von der Schlaggeschwindigkeit u und der Schlagamplitude d/2 abhängt.

Von großer praktischer Bedeutung ist nun die weitere Feststellung, daß eine ähnliche Wirbelschleppe fast genau gleichen Wirbelflusses nach (9) auch zustande kommt, wenn der Flügel im Stand, also bei v = o schlägt. Wegen der Abrundung der Nase des Profils bildet sich wieder vorwiegend nur der Anfahrwirbel am scharfen Schwanzende. Die Schleppe der jetzt viel enger gruppierten Wirbel bewegt sich durch ihre Eigengeschwindigkeit w_a gemäß Gl. 18 einseitig vom Schwanzende (nach rechts) fort.

Weiterhin ist praktisch wichtig, daß wegen der einseitigen Orientierung des Schwanzendes ein gegenläufiger Strahl (nach links) niemals zustande kommen kann. (Fische können mittels der Schwanzflosse niemals rückwärts schwimmen).

c) Kraft und Leistungsbetrachtungen.

Mit der soeben gewonnenen Anschauung des Strömungsbildes und der quantitativen Ermittlung des Wirbelflusses können nun die Kraftwerte für den Schub sofort angegeben werden. Durch die soeben gemachten (nicht schwerwiegenden) Voraussetzungen entsteht genau das Bild eines nichtstationären, aber periodischen Idealpropellers gemäß Abschn. 2, so daß alle dort gewonnenen Formeln hier anwendbar sind.

Gemäß Gl. 3), 5) und 9) ist der mittlere Schraubenschub

$$S_m = \Delta p_m \cdot F = \varrho \frac{\Gamma \cdot u}{d} \cdot d \cdot 1$$

Für die Propellerfläche als Strahlfläche ist hierbei für den laufenden Meter der Flügellänge die Fläche $F = d \cdot 1$ einzusetzen, da an deren Berandung der Wirbelfluß entsteht.

$$\text{damit } S = \varrho \cdot \Gamma \cdot u \quad \ldots\ldots \quad (10)$$

Es entsteht ferner ein Schraubenstrahl (alles wie beim Idealpropeller) der mittleren Axialgeschwindigkeit c_a

$$\text{wobei } \left(v + \frac{c_a}{2}\right) c_a = \left(\frac{d\Gamma}{dt}\right)_m = \frac{\Gamma u}{d} = \frac{S_m}{\varrho \cdot d} \text{ ist}$$

Dieser Schraubenstrahl hat sich je nach der Belastung kontrahiert. Am Ort der Propellerfläche induziert die Wirbelschleppe die mittlere Axialgeschwindigkeit $c_a/2$.

Der bestmögliche Propulsionswirkungsgrad des Schlagflügels ist

$$\eta_{id} = \frac{2}{2 + \frac{c_a}{v}} \quad \ldots\ldots \quad (11)$$

(je größer die Schlagamplitude, desto kleiner c_a und größer η_{id}).

Diese einfachen Ergebnisse werden zweckmäßig durch eine Kraftbetrachtung am Propellerflügel selbst, wo ja der Schub nur entstehen kann, kontrolliert.

Das Anströmungsdreieck des Bildes 2 unter Anwendung des Kutta-Joukovski-Satzes ergibt bei der am Flügel vorhandenen Zirkulation Γ jeweils stets für die Längeneinheit des Schlagflügels

den Auftrieb $A = \varrho \cdot \Gamma \cdot w$

die Schubkomponente $S = A\, u/w = \varrho \cdot \Gamma \cdot u \quad (10\,a)$

und die Querkomponente $T = A\left(v + \frac{c_a}{2}\right)/w = \varrho\, \Gamma \left(v + \frac{c_a}{2}\right)$

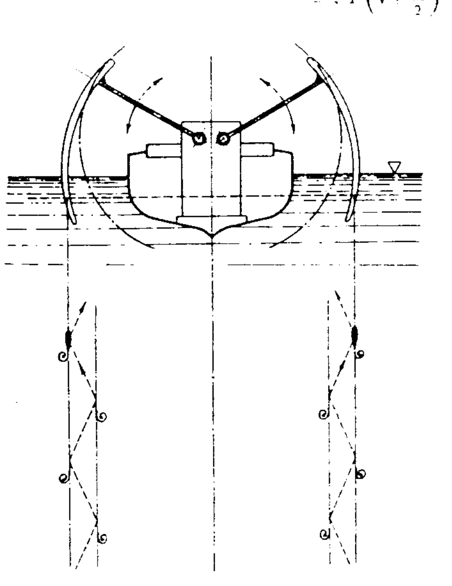


Abb. 3. Prinzip eines Schlagflügelpropellers mit beiderseitig schwingenden, gebogenen Armen (Nach Ing. Budig, Dingelsdorf.)

der Wirkungsgrad ist:

$$\eta_{id} = \frac{S \cdot v}{T \cdot u} = \frac{\varrho \cdot \Gamma u \cdot v}{\varrho \cdot \Gamma \cdot \left(v + \frac{c_a}{2}\right) \cdot u} = \frac{2}{2 + c_a/v} \quad (11\,a)$$

Die Gleichungen (10a) und (11a) sind mit 10 und 11 identisch.

Ein weiterer, hier nicht näher zu erörternder Verlust kommt durch die in Bild 2 ersichtliche Neigung der Grundströmung (infolge Induktion von Quergeschwindigkeiten c_u am Flügelort) zustande, wodurch bei gleichem Auftrieb und annähernd gleicher Querkraft eine Verringerung der Schubkomponente entsteht.

d) **Anwendungen.**

Das Schlagflügelantriebsprinzip liegt einer Konstruktion des Ing. Budig, Dingeldorf (Bodensee) zugrunde. Die konstruktive Anordnung zeigt Bild 3, eine praktische Ausführung an einem Faltboot die Bilder 4) und 5). Die beiderseits auf- und abschwingenden Flügel bewirken, wenn man den Vorgang auf einer Ebene gleicher Wassertiefe beobachtet, eine schlagende periodische Querbewegung der beiden Profile gemäß dem oben erörterten Typ. Die in Bild 3 und 2 eingezeichneten Wirbelschleppen konnten deutlich in der Natur beobachtet werden. Das Boot fährt gut aus dem Stand an und erreicht, durch einen Mann angetrieben, eine höhere Geschwindigkeit als ein leichteres Boot mit 2 Paddlern. Allerdings kann es mit dem Propeller nicht stoppen oder reversieren.

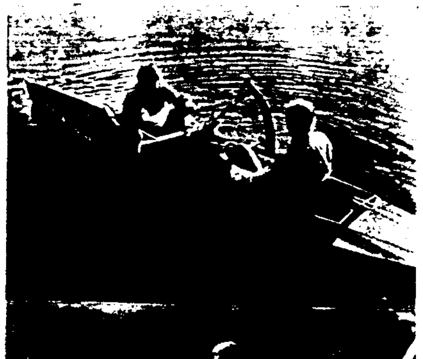


Abb. 4. Ausführung von Abb. 3, Schlagflügel in Arbeitsstellung. (Foto: Dürr-Firmenich, Karlsruhe-Durlach.)

Abb. 5. Wir Abb. 4, die Flügel sind zum Transport hochgestellt. (Foto: Dürr-Firmenich, Karlsruhe-Durlach.)

Bild 6) zeigt einige Bilder von Fischen. Durch einfaches Schlagen des Schwanzes reißen an dessen stets scharfer Hinterkante Anfahrwirbel ab, die gleichfalls zu einem Schlagflügelantrieb führen. Kräftige, beiderseits oberhalb der Wirbelsäule gelegene, bis zum Kopf reichende Muskelstränge bewirken die Schwanzbewegung. Die übrigen Flossen dienen i. a. nur zum Steuern oder Rückwärts-Schwimmen oder zur Vergrößerung der meist stark betonten Lateralfläche. Die Vortriebsbewegung des Schwanzes zusammen mit dem profilierten Rumpf kann eine Wirkung nach Art des Flettner-Ruders ergeben (besonders bei Macropoden und Plattfischen); ungleichförmige Schläge rufen sehr wirksame Steuerkräfte hervor.

Die Seeleuten geläufige Beobachtung, daß die Schnelligkeit von Fischen und Delphinen die von Schiffen weit übertreffen kann, beweist die technische Güte des Schlagflügelantriebs.

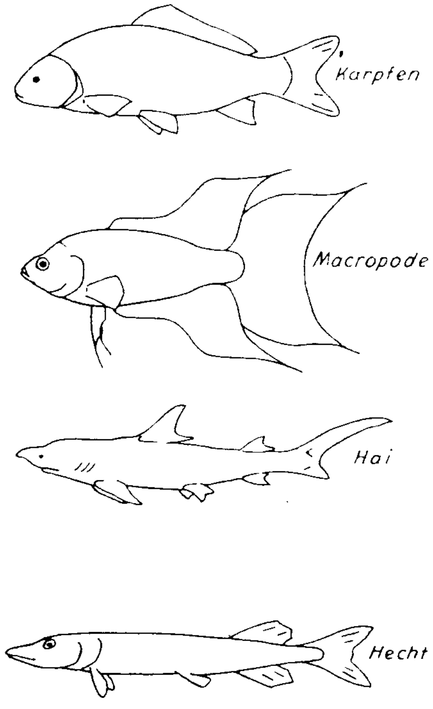


Abb. 6. Verschiedene Formen von Fischen.

4) **Der Stoßpropeller**

a) **Der Propellermechanismus.**

Das jetzt zu beschreibende Vortriebsorgan soll „Stoßpropeller“ genannt werden, da die Propulsion durch den periodischen Ausstoß von Wasser aus einem nach rückwärts offenen Rohr mit beliebigem Querschnitt entsteht. Eine solche Anordnung ist m. W. bisher nur bei einem vortrefflich erdachten Spielzeugboot angewendet worden. Nachstehende qualitative und quantitative Untersuchung wird zeigen, daß der Stoßpropeller — bei Einhaltung gewisser notwendiger Bedingungen — ein vollwertiger Propeller ist, der u. U. besonders einfache Maschinenanlagen ermöglichen kann.

Die grundsätzliche Wirkungsweise ist etwa aus Bild 7 zu ersehen.

In einem nach hinten offenen, mit scharfer Kante versehenen Rohr bewegt sich eine Wassersäule — etwa durch Wirkung eines Kolbens — periodisch hin und her. Dies ist bereits alles, und es ist nicht ohne weiteres verständlich, wie hierdurch ein Vortrieb — noch dazu normal hohen Wirkungsgrades — zustandekommen soll. Die Wirkung wird aber sofort ersichtlich, wenn man sich vergegenwärtigt, daß im Ansaugetakt das Wasser nach Art einer Senke aus dem gesamten Flüssigkeitsraum (Absolutströmung) angesaugt wird, während beim Ausstoßen durch das Abreißen der Strömung an der scharfen! Hinterkante des Rohres ein ausgeprägter Strahl nur nach rückwärts geworfen wird, wie der Rauchaustritt aus Schornsteinen. Es herrschen also insoweit gleiche Verhältnisse wie bei Schraubenpropellern, nur daß dort das gleichzeitige Ansaugen auch einen Schubanteil liefert, während es hier scheinbar in falscher Richtung am Kolben zieht. Der Vorgang bedarf also eingehenderer Aufklärung.

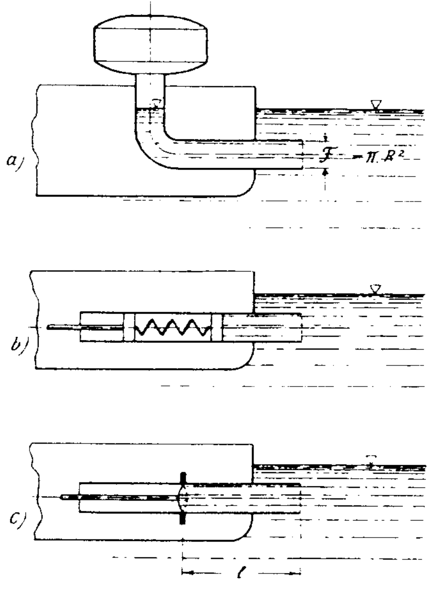


Abb. 7. Prinzip des Stoßpropellers. — Die Wassersäulen sind schwingfähig durch: a) Erdanziehung, b) Federkraft, c) Kompressibilität des Wassers selbst.

b) **Das Strömungsfeld beim Standpropeller.**

Die wesentlichen Merkmale des eigenartigen, nicht stationären Strömungsfeldes, wie es durch die Wirkung des Stoßpropellers entsteht, lassen sich am besten für den Standpropeller erläutern, wobei zunächst an das ebene Problem gedacht wird, bei dem zwei senkrechte Wände das „Rohr“ bilden.

Stationäres Ansaugen (siehe Bild 8).

Die im völlig reibungsfreien Feld, wie es im Anhang C erläutert ist, sich einstellende Senkenströmung im Außenbereich bildet sich auch hier — mit Ausnahme der mündungsnächsten Stellen — heraus. An der scharfen Kante der Mündung entstehen jedoch infolge der dort im Anfahren vorhandenen hohen Geschwindigkeitsgradienten durch Zähigkeitswirkung freie Wirbel, deren Induktion gerade das unendlich schnelle Umströmen der scharfen Kante verhindert. Hierdurch wird die Strömung, wie gezeichnet, zunächst zur Mitte des Rohres abgelenkt, wie dies auch bekannt ist vom Ausfluß aus einer Bordamündung [9]. Bewegt sich ein ansaugender Kolben mit der gleichmäßigen Geschwindigkeit c_k nach links, so muß aber aus Gründen der Kontinuität im Bereich vor ihm auch die mittlere Strömungsgeschwindigkeit c_k herrschen, so daß sich die durch die Strahleinschnürung an der Eintrittskante bildende Übergeschwindigkeit wieder ausgleichen muß.

Für das Ansaugen wäre deshalb eine Abrundung an der Mündung verlustfreier und deshalb besser.

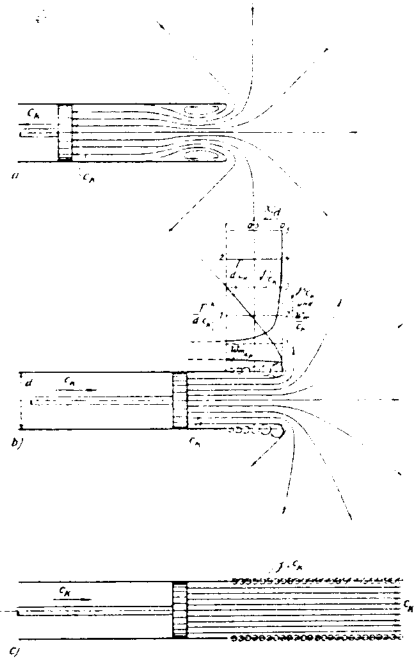


Abb. 8. Prinzipskizze des stationären Ansaugens und Ausstoßens aus Rohren: a) Dauerndes Ansaugen, b) Beginn des Ausstoßens, c) Dauerndes Ausstoßen.

Stationäres Ausstoßen (siehe Bild 9).

Wir gehen wieder von einem völlig ruhenden Flüssigkeitsfeld aus und nehmen an, daß der Kolben plötzlich nach rechts drückend mit konstanter Geschwindigkeit c_k bewegt wird. Zunächst stellt sich die im Anhang C erörterte wirbelfreie Potentialströmung, jetzt jedoch als Qellströmung mit nach außen gerichteten Geschwindigkeiten ein mit den unendlich großen Geschwindigkeitsgradienten an der scharfen Kante. Dort entstehen daher Wirbel in solcher Stärke und Drehrichtung (vgl. Bild 8b), daß ihre Induktion eine Strömung quer zur Kante gerade verhindert. Diese kontinuierlich entstehenden freien Wirbel werden nun durch die induzierende Wirkung der anderen Rohrseite von der Mündung fortgeschwemmt. Eine genauere Prüfung zeigt folgendes Bild: Die eben erwähnte Geschwindigkeit der frei entstehenden Wirbel ist identisch mit der im Anhang C, vgl. Abb. 21, angegebenen mittleren Geschwindigkeit w_m, sie ist etwas kleiner als $c_k/2$ und wird bei länger

$c_k/2$. Da diese Gruppengeschwindigkeit annähernd konstant ist, kann weiterhin gefolgert werden, daß die entstandenen freien Wirbel ihre gegenseitige Gruppierung annähernd (gut) beibehalten. Da nun die in Abb. 21 gezeigte und in Abb. 9 übertragene Zirkulationsbelegung γ gerade die Grenzbedingung verschwindender Querströmung erfüllt, folgt unter obiger Voraussetzung, daß diese γ-Kurve uns jetzt auch ein recht genaues Bild der Verteilung der freien Wirbel hinter der Abstromkante gibt. Diese Kurve zeigt uns dann in jedem Zeitpunkt folgendes:

Erstens: In ihrem, in Abb. 9 als ausgezogene Linie rechts von der Mündung gelegenen Teil die Stärke der physikalisch wirklich existierenden, sichtbaren (vgl. Bild 11a und b) Wirbelschicht und

Zweitens: In dem links von der Kante gelegenen gestrichelten Teil die Stärke der (gedachten) Stabwirbelbelegung wie in Anhang C.

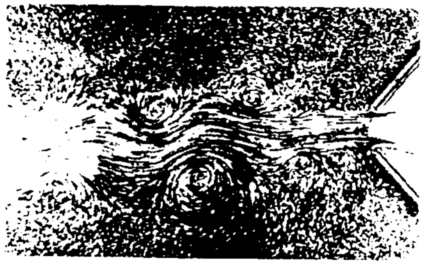


Abb. 9. Lichtbild eines stationär ausgestoßenen Strahles
Aufnahme: Institut Föttinger, Berlin

Während also der Kolben das Wasser mit c_k vor sich herschiebt, wandert die Wirbelbelegungskurve mit $w_m \cong c_k/2$ in gleicher Richtung. Die dementsprechend hinter der Mündungskante entstehenden Potentialwirbel pressen das mit c_k strömende Rohrwasser in ihrem Bereich noch genau und solange in die Rohrrichtung, bis sie schließlich an ihrem rechten Ende das überschüssige Wasser quellenartig in den ganzen Flüssigkeitsraum entlassen. Die sekündliche Wirbelerzeugung an der Hinterkante wird gemessen durch den Wert $\gamma \cdot w_m = d\Gamma/dt$. Wenn die Wirbelschicht in einem bestimmten Zeitpunkt t bis zu einem Abstand x fortgeschwemmt wurde, läßt sich die bis dahin insgesamt erzeugte Wirbelstärke Γ aus der Fläche der Belegungskurve $\Gamma = \int_0 \gamma \cdot dx$ ermitteln, wobei $x \cong w_m \cdot t$ ist. Dieser Wert ist in 8b gleichfalls eingetragen. Nach längerer Zeit wird der Wirbelfluß genau

$$\frac{c_K^2}{2} = \frac{d\Gamma}{dt} \qquad \ldots\ldots \qquad (12)$$

Beim Beginn des Ausstoßens ist der Wirbelfluß jedoch größer (er beginnt sogar unendlich). Wir benötigen ferner noch einen zeitlich nach Gl. 2) gemittelten Wert:

$$\left(\frac{d\Gamma}{dt}\right)_m = \frac{1}{t}\int_0^t \gamma \cdot w_m \cdot dt = \beta \cdot c_K^2 \qquad \ldots\ldots \qquad (12a)$$

der uns den zeitlichen Mittelwert des Wirbelflusses bis zum Zeitpunkt t bzw. bis zu dem zugehörigen Abstand x des Anfangs der Wirbelschleppe bis zur Mündung angibt. Der Faktor β ist aus Bild 21b zu entnehmen, er wird nach einiger Zeit des Ausstoßens $^1/_2$ nach Gl. 12.

Erfolgt nun der Ausstoß lange genug, so wandert die Wirbelschleppe immer weiter in den Abstrom und schafft so einen langen, gerichteten Strahl der konstanten Innengeschwindigkeit c_k. Siehe das Phot in Abb. 9. Aus letzterem wird ersichtlich, daß solche geraden Wirbelschichten labil sind und das Bestreben zeigen, sich zu Einzelwirbeln aufzurollen [10], ohne jedoch noch Rückwirkungen auf dae Ausstoßkante auszuüben. Dieser Vorgang kann daher hier mit Recht vernachlässigt werden.

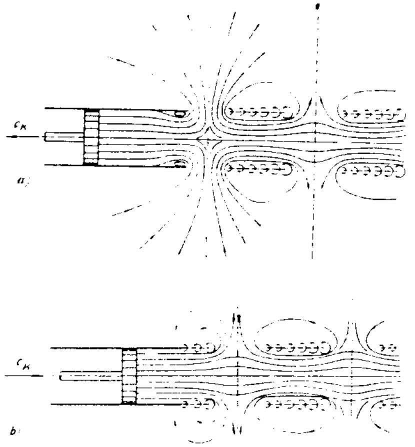


Abb. 10. Prinzip des periodisch arbeitenden Stoßpropellers: a) Ansaugetakt, b) Stoßtakt.

Die vorstehend für ebene, d. h. durch 2 Ebenen begrenzte Mündungen abgeleiteten Ergebnisse gelten qualitativ und weitgehend auch quantitativ für runde Rohre. Es entstehen hier Wirbelringschichten, wobei der Wirbelfluß ähnlich wie in (12a) gleichfalls durch den Wert $(d\Gamma/dt)_m = \beta \cdot c_k^2$ angebbar ist. Der Faktor β strebt gleichfalls für lange Wirbelschleppen (großen Kolbenhub) gegen den Wert $^1/_2$, wird aber nach Überlegungen in [7] für kurze Schleppen bei Ringwirbeln beträchtlich größer als bei Stabwirbeln.

Periodisches Ansaugen und Ausstoßen, siehe Bild 10.

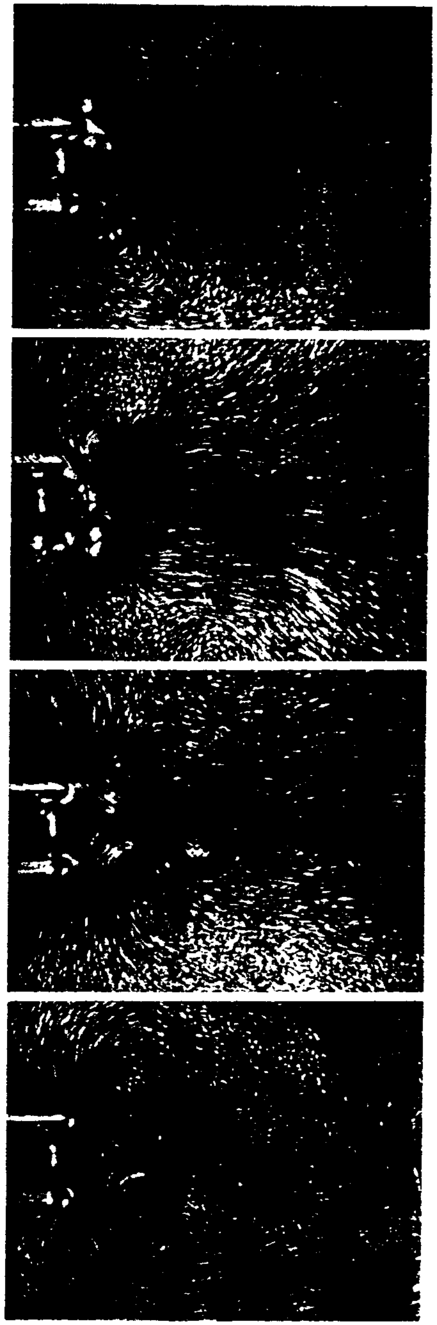
Die Flüssigkeit im Rohr bewege sich, etwa durch Wirkung eines Kolbens, periodisch (Vollperiode T) hin und her, wobei wir annehmen wollen, daß im Ansaugetakt $T/2$ die konstante Kolbengeschwindigkeit c_k nach links und im Ausstoßtakt $T/2$ gleichfalls die Geschwindigkeit c_k, aber nach rechts herrsche.

Beginnen wir mit einem Ausstoßtakt, so entsteht eine Wirbelschicht hinter der Mündung, wie soeben beschrieben (Abb. 9), mit dem mittleren Wirbelfluß

$$\left(\frac{d\Gamma}{dt}\right)_{T/2} = \beta \cdot c_k^2 \qquad \text{nach Gleichung 12a}$$

mit einer Gesamtzirkulation $\Gamma = \beta \cdot c_k^2 \cdot \frac{T}{2}$ (12b)

Am Schluß des Taktes bleibt der Kolben stehen, der flächige Ringwirbel wird aber durch Eigeninduktion stromabwärts weiterschwimmen. Mit Ausnahme der mündungsnächsten Teile wird er diese Bewegung auch (wegen der sehr schnell abklingenden Senkenwirkung) angenähert fortsetzen, wenn nun der Ansaugetakt beginnt. Das in das Rohr jetzt angesaugte Wasser kommt aus dem gesamten Außenraum und wird hauptsächlich und nach weiteren Takten ausschließlich durch die jetzt wirbelfreie zylindrische Ringfläche zwischen Mündung und dem nächstgelegenen Stoßwirbel durchgezogen, siehe Bild 10a.



Unterschriften in der Reihenfolge der links abgebildeten Schaumfotos: Abb. 11. Lichtbilder eines periodisch arbeitenden, ebenen Stoßpropellers: Beginn des Stoßtaktes, Mitte des Stoßtaktes, Beginn des Ansaugetaktes, Mitte des Ansaugetaktes.

(Versuche und Aufnahmen: K. Trunz, Karlsruhe)

Bei fortgesetzter Wiederholung der Hin- und Herbewegung entstehen schließlich Strömungsbilder, wie sie in Bild 10a und 10b dargestellt sind mit einem ausgeprägten Propellerstrahl, der aus periodisch sich folgenden, kürzeren Ringwirbelschichten der Stärke Γ nach Gl. 12b gebildet wird. Während einer Periode T wird also — nur im Ausstoßtakt — eine bleibende Wirbelstärke erzeugt, die zu folgendem, zeitlich gemittelten Wirbelfluß führt:

$$\left(\frac{d\Gamma}{dt}\right)_m = \frac{\Gamma}{T} = \beta \cdot \frac{c_K^2}{2} \qquad \ldots\ldots \qquad (13)$$

Bemerkenswert ist weiterhin, daß dieser Strahl durch Eigeninduktion auch während des entgegengerichteten Ansaugetaktes selbst Wasser aus dem Gesamtraum ansaugt und damit den Strahl kontinuierlich beschickt. Dieser Effekt ist aus den beiden letzten Fotos auf Abb. 11 besonders deutlich zu sehen. Diese Wirbelaufnahmen eines ebenen Stoßpropellers sind mit einer Versuchseinrichtung nach Bild 12 gewonnen worden. Die freie Oberfläche des Wassers war mit Aluminiumpulver bestreut.

c) Kraftverhältnisse beim Standpropeller.

Das in b) erläuterte Strömungsfeld führt uns nun wieder auf den Vorgang eines in Abschn. 2 behandelten instationären Idealpropellers. Wir entnehmen vorstehenden Ableitungen lediglich die eine Tatsache, daß dort mit einer Periodizität T Wirbelringe einer uns (gemäß Gl. 12b bzw. 13) bekannten Intensität Γ entstehen, die einen Strahl bilden. In einer Kontrollebene weit hinter dem Propeller würde man einen periodischen Wirbelfluß feststellen, der im zeitlichen Mittel für die Vollperiode mit nur einem Stoßakt beträgt:

$$\left(\frac{d\Gamma}{dt}\right)_m = \frac{\Gamma}{T} = \beta \cdot \frac{c_K^2}{2} \qquad \ldots\ldots \qquad (\text{wie Gl. 13})$$

Gl. 3 in Abschn. 2 lehrte uns, daß in solchem Falle am Ort der Wirbelentstehung ein (zeitlich gemittelter) schuberzeugender Drucksprung Δp_m nach Gl. 2 vorhanden sein muß.

$$\Delta p_m = \varrho\left(\frac{d\Gamma}{dt}\right)_m = \varrho \cdot \frac{\beta}{2}\, c_K^2$$

und ein mittlerer Propellerschub

$$S_m = \Delta p_m \cdot F = \varrho/2\, F \cdot c_K^2 \cdot \beta \qquad \ldots\ldots \qquad (14)$$

Die axiale mittlere Zuwachsgeschwindigkeit im Strahl, der im Stand auf $F/2$ kontrahiert ist, beträgt nach Gl. 4)

$$\frac{c_a^2}{2} = \left(\frac{d\Gamma}{dt}\right)_m = \frac{\beta}{2}\, c_K^2$$

Der im Ergebnis erscheinende Faktor β (vgl. Bild 21b) kann sehr verschiedene Größen haben. Er ist mindestens $= {}^1/_2$, kann aber — abhängig von der Hubhöhe, der Form des Rohrquerschnitts und der Bewegungsgleichung des Wassers im Rohr — sehr viel größer werden. Er wird entweder durch Lösung von Integralgleichungen oder durch Versuche bestimmt.

Aufschlußreich ist es noch, das Volumen der kontinuierlich im Strahl bewegten Wassermenge zu errechnen:

$$Q = \frac{F}{2} c_a = F \cdot \frac{1\,\beta}{2} c_K \quad \cdots\cdots\cdots\cdots (15)$$

Vergleicht man diesen Wert mit derjenigen Wassermenge, die im Rohr nur während des Ansaugetaktes gesammelt und dann direkt in den Strahl ausgestoßen wird:

$$Q_s = F \cdot c_K \cdot \frac{T/2}{T} = \frac{1}{2} F \cdot c_K$$

so wird ersichtlich, daß Gl. 15 größere Werte annehmen kann als diesen, sofern $\beta > 1$.

Diese Feststeellung wird durch die Strömungsbilder 10a und 11c und d bestätigt und erklärt, nach denen auch im Ansaugetakt frisches Wasser in den Strahl fließt.

Um diese merkwürdigen Ergebnisse auch quantitativ zu überprüfen, wurden Versuche mit einer Apparatur nach Bild 13 gemacht. Hierbei bewegte sich ein Kolben sinusförmig mit verschiedenen Geschwindigkeiten in einem vertikalen Glasrohr auf und ab. Der mittlere Propellerschub wure nicht am Kolben, wo er ja wirklich auftritt, sondern durch die Reaktionskraft auf das große Wassergefäß gemessen, das hierzu auf eine empfindliche Wage gestellt war. Die Meßergebnisse sind in Bild 14 aufgetragen, und zwar: der Schub in gr über der mittleren Kolbengeschwindigkeit c_k

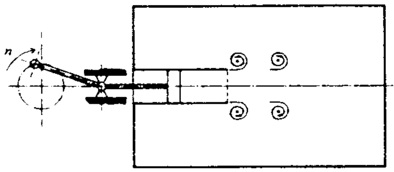


Abb. 12 Versuchseinrichtung für die Wirbelaufnahmen des Stoßpropellers von Abb. 11. Die Wasserbewegung wird durch auf die Oberfläche gestreutes Aluminiumpulver sichtbar gemacht.

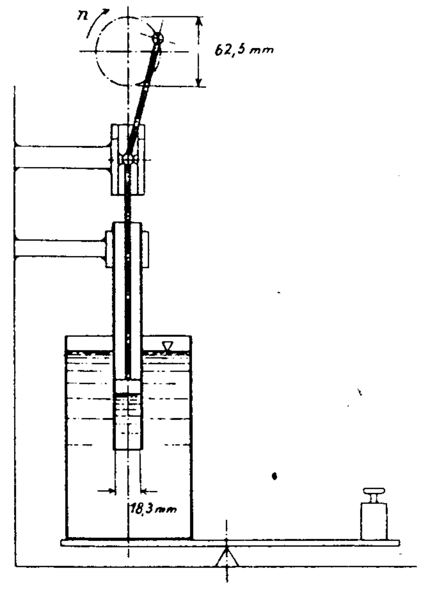


Abb. 13 Prinzip einer Versuchsanordnung zur Schubmessung. Der wirksame mittlere Schub wird als Reaktion am Wassergefäß gewogen. (Nach Vorschlag E. G. Finke, Karlsruhe.)

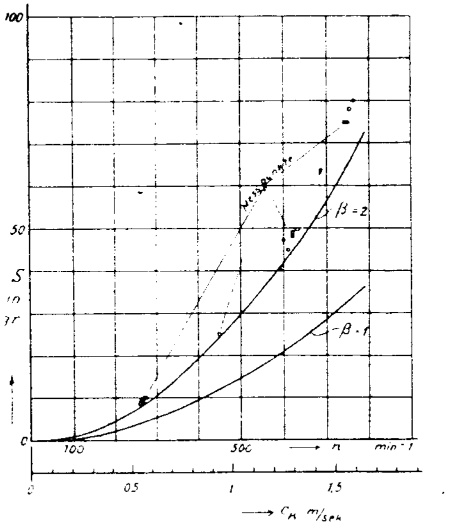


Abb. 14 Versuchsergebnisse mit der Anordnung von Abb. 13. Meßpunkte sind als Kreise eingezeichnet. Ausgezogene Linien sind theoretische Ergebnisse nach Gl. 14. Für den Faktor β siehe Gl. 12a und Abb. 21.

$c_k = \frac{2}{\pi}$ Geschwindigkeitsamplitude = prop. Drehzahl.

In dasselbe Schaubild ist der theoretisch ermittelte Schub nach der Aussage von Gl. 14 als Kurve eingetragen, wobei für β die Werte 1 und 2 angenommen wurden.

Die Übereinstimmung ist für $\beta = 2$ recht befriedigend, insbesondere kommt die quadratische Abhängigkeit von der mittleren Kolbengeschwindigkeit klar heraus.

d) Leistungsbetrachtungen.

Der in Gl. 14) aus Wirbel-Impulsbetrachtungen gewonnene mittlere Schub kann nur durch Überdruckwirkung im Rohr des Stoßpropellers, etwa an einem dort die notwendige Wasserbewegung bewirkenden Kolben, entstehen, und zwar während des Stoßtaktes. Dies schließt jedoch nicht aus, daß sich diesem wirksamen Überdruck noch zusätzliche, mit dem Takt wechselnde Unter- und Überdrucke überlagern, die sich in ihrer Größe gegenseitig zwar aufheben, die aber doch notwendig vorhanden sein müssen, um das Wasser anzusaugen und um die Trägheitwiderstände des hin- und herbewegten Wassers zu überwinden. Diese periodisch wechselnden, keinen Beitrag zum Schub liefernden Drücke würden den Wirkungsgrad des Stoßpropellers prohibitiv herabsetzen, wenn sie unter Zufuhr von Maschinenenergie durch einen Kolben aufgebracht werden müßten.

Hier kann jedoch ein wesentlicher, konstruktiver Gedanke Abhilfe schaffen. Es ist durchaus möglich, eine in einem Rohr befindliche Wassermenge durch Einschaltung eines elastischen Bauelements (Feder, Erdschwere usw.) zu einem schwingungsfähigen System zu machen, das bei sonstiger Verlustfreiheit ohne Energiezufuhr hin und her schwingen kann. Im Anhang D sind einige solcher schwingungsfähiger Anordnungen in Abb. 22 zusammengestellt vgl. auch Abb. 7. Die notwendige Energiezufuhr durch eine Maschinenanlage hat dann aber, wenn Synchronismus mit der Eigenschwingungsfrequenz der Wassersäule besteht, nur die Dämpfungswiderstände zu überwinden, und diese werden hier hauptsächlichsächlich durch die Wirbelablösung an der scharfen Rohraustrittskante verursacht. Das ist ja aber gerade das Äquivalent für die Schraubenschubleistung. Von den verschiedenen sich hier bietenden konstruktiven Möglichkeiten sind in den Bildern 7 und 22 nur wenige schematisch angedeutet.

Beachtung dürfte nach Bild 7a eine Anordnung verdienen, in der unter Vermeidung einer normalen Maschinenanlage in dem angedeuteten Zylinder am inneren Ende des Schubrohres ein direkt auf das Wasser drückender und leicht zu steuernder Verbrennungsgasdruck nach Art einer Humphreypumpe [11] und [12] wirken könnte. Die Einwirkung der Maschinenleistung muß natürlich in allen Fällen mit 90° Phasenverschiebung erfolgen.

Ein ähnliches Prinzip ist in einem kleinen Modellboot nach Bild 15 verwirklicht. Die 2 Stoßrohre enden innen in einem flachen, oben durch eine Membran abgeschlossenen Kessel. Durch Wirkung einer Flamme wird hier Dampf erzeugt, der das erhitzte Wasser aus den Rohren treibt. Beim Zurückschwingen wird kühles Wasser von allen Seiten angesaugt, das dann erneut verdampft. In einem durch die Eigenschwingungsperiode der Rohre gesteuerten Rythmus der Schwingungen, deren Amplitude durch die Wärmezufuhr bestimmt wird, bewegt sich das Modellboot gut vorwärts. In heißem Wasser schwimmend kommt der Schwingungsvorgang nicht zustande. Nur theoretisches Interesse dürfte eine Anordnung nach Bild 7c haben, bei der die Elastizität des Wassers als Federkraft verwendet wird (Schallschwingungen). Ob sich die notwendigen Stoßwirbel in so kurzer Zeit bilden können, wäre noch zu prüfen.

Zusammenfassend kann bei Verwendung schwingungsfähiger Systeme festgestellt werden, daß als Verlustleistungen in der Hauptsache nur zwei auftreten werden:

Der normale, bei jedem Propeller entstehende, nur vom Schubbelastungsgrad abhängige kinetische Austrittsverlust, der beim Stoßpropeller durch günstige Ausbildung der Strahlfläche gering gehalten werden kann. Ferner der Verlust durch Wandreibung, der hier wegen relativ großer benetzter Flächen noch sorgfältig studiert werden müßte. Andererseits führen pulsierende Bewegungen bekanntlich zu geringen Reibungsbeiwerten. Es muß weiterhin bedacht werden, daß Stoßpropeller genau wie Schlagflügelpropeller nicht rückwärts arbeiten können, daß also zum Stoppen und Reversieren wie bei Fischen besondere Einrichtungen erforderlich sind.

e) Verhalten in Fahrt.

An der Propellermündung entstehen beim Ausstoßen strahlbildende Randwirbel erst dann, wenn die Kolbengeschwindigkeit c_k die Fahrtgeschwindigkeit v übersteigt. Die mittlere Wirbeldichte des Abstroms im Stoßtakt ist

$$\gamma = 2 \cdot \beta \cdot (c_k - v)$$

Die erzeugten Wirbel fließen hier mit der Geschwindigkeit

$$w_m \sim v + \frac{c_K - v}{2} = \frac{v + c_K}{2} \text{ ab.}$$

Der Wirbelfluß während des Stoßtaktes ist also

$$\left(\frac{d\Gamma}{dt}\right)_T = \gamma \cdot w_m \sim \beta\,(c_K - v)(c_K + v) = \beta\,(c_K^2 - v^2)$$

sowie gemittelt nach Gl. 2 für eine volle Periode

$$\left(\frac{d\Gamma}{dt}\right)_m \sim \frac{1}{2}(c_K^2 - v^2)\cdot\beta$$

dies ergibt den Schraubenschub nach Gl. 3 und 5).

$$S_m = \rho \cdot F \left(\frac{d\Gamma}{dt}\right)_m \sim \frac{\rho}{2} F (c_K^2 - v^2)\,\beta = \rho\, F\, c_a \left(v + \frac{c_a}{2}\right)$$

Hieraus errechnet sich angenähert die notwendige Kolbengeschwindigkeit c_k bei vorgegebenem Schub S. Versuche hierzu sind bisher nicht durchgeführt worden.

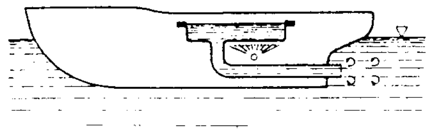


Abb. 15. Prinzip eines Modellbootes mit Stoßpropellern und Dampfantrieb

5) Zusammenfassung.

Die Arbeitsweise zweier nicht stationär aber periodisch arbeitender Vortriebsorgane wird untersucht. Durch Ermittlung von notwendigerweise entstehenden Wirbelschleppen wird einerseits das Bild der entstehenden Strömungen beschrieben, andererseits aber unter Verwendung eines hergeleiteten einfachen Impulssatzartigen Gesetzes über den Wirbelfluß die Größe der wirksamen Schubkraft bestimmt.

Die beiden untersuchten Propellerarten erweisen sich als durchaus hochwertig, sie sind jedoch nicht ohne weiteres umsteuerbar.

Für Sonderfälle könnte daher die Anwendung dieser neuen instationären Propeller durchaus erwogen werden.

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Anhang.

A) Freie Wirbelschichten, Wirbelreihen auf Ebenen oder Zylinderflächen.

A1) Die unendlich ausgedehnte Ebene ($x = 0$), gleichmäßig belegt mit Stabwirbeln senkrecht zur x-y-Ebene der konstanten Belegungsdichte

$$\gamma \left(\text{in } \frac{m^2}{\text{sek}\; m} = m\ \text{sek}\right) \qquad \text{Siehe Bild 16a}$$

In einem beliebigen Aufpunkt (x, y) errechnet sich die Geschwindigkeit zu $c_y = 0$ aus Symmetriegründen:

$$c_x = \mp c\,\frac{j \cdot y}{2\pi}\int_{-\infty}^{+\infty}\frac{d\,\xi}{(x-\xi)^2+y^2} = \begin{cases} +\gamma : \text{für } y > 0 \\ -\gamma : \text{für } y < 0 \end{cases}$$

nach dem Biotsavartschen Gesetz mit einem unstetigen Geschwindigkeitssprung an der Wirbelschicht der Größe γ. Letztere Feststellung gilt auch für alle beliebig geformten Wirbelschichten mit beliebig veränderlicher, aber stetiger Belegung $\gamma(\xi)$.

A2) Zwei parallele, ebene, unendlich lange Wirbelschichten wie in A1) im Abstande d, jedoch mit gegenläufiger Wirbelbelegung γ, siehe Bild 16b.

Geschwindigkeitsverteilung durch Superposition der Ergebnisse von A1) c_y überall Null, c_x im Außenraume oberhalb und unterhalb der Wirbelschichten null.

c_x zwischen beiden Wirbelschichten: $c_x = c = \gamma$ mit je einem unstetigen Geschwindigkeitssprung von $+\gamma$ bzw. $-\gamma$ an der oberen und unteren Schicht.

Da die einzelnen Wirbelfäden ($\gamma\,d\xi$) nach den Helmholtzschen Wirbelsätzen mit der Strömung fortgeschwemmt werden, ist ihre mittlere, nach rechts gerichtete „Gruppengeschwindigkeit" $w_m = \gamma/2 = c/2$ gegenüber dem im Unendlichen ruhenden Raume, und zwar für die obere Wirbelreihe durch die Induktion der unteren Wirbelreihe nach Ziffer A1 und umgekehrt.

Eine fest im absoluten (ruhenden) Bezugssystem liegende Kontrollfläche wird daher durchflossen mit dem Wirbelfluß

$$\left(\frac{d\Gamma}{dt}\right)_{abs} = \gamma \cdot w_m = \frac{\gamma^2}{2} = \frac{c^2}{2} \quad \ldots\ldots\ldots \quad (16)$$

Eine mit der Fahrtgeschwindigkeit v nach links, d. h. entgegen dem Strahl bewegte Kontrollfläche wird durchflossen mit dem Wirbelfluß

$$\left(\frac{d\Gamma}{dt}\right)_v = \gamma(v + w_m) = \gamma(v + \gamma/2) = \gamma\left(v + \frac{c}{2}\right) \quad (16a)$$

Sekündlich die vorgenannten Kontrollflächen (nach rechts) passierende Wassermasse:

$$\varrho \cdot Q_{abs} = \varrho \cdot F \cdot c = \varrho \cdot d \cdot 1 \cdot c \quad \ldots\ldots\ldots \quad (17)$$

$$\text{bzw.} \quad \varrho \cdot Q_v = \varrho \cdot F(v+c) = \varrho \cdot d \cdot 1\,(v+c) \ldots\ldots \quad (17a)$$

Je zwei sich auf den Wirbelflächen gegenüberliegende Wirbelteile $\gamma\left(\frac{+d}{2}\right)\cdot d\,\xi$ und $\left|-\gamma\left(-\frac{d}{2}\right)\cdot d\,\xi\right|$ schließen schließen sich im Unendlichen [$z \to \pm\infty$] mit Beibehaltung des Drehsinns und bilden so eine geschlossene Wirbelfigur nach den Ausführungen des 2. Hauptkapitels bei Anwendung auf ebene Probleme (Strömung in allen x y-Ebenen identisch).

A3) Unendlich lange Wirbelschichten mit konstanter Belegung γ auf beliebig zylindrischen Flächen, insbesondere auf Kreisflächen des Durchmessers d.

Die Achsen der geschlossenen Wirbellinien $\gamma d\xi$ liegen alle auf achsennormalen Ebenen.

Es gelten die gleichen Formeln für Geschwindigkeiten und Wirbelfluß wie im vorigen Abschnitt A2.

Bild 16b kann z. B. auch als koaxialer Zylinderschnitt einer Kreiswirbelschicht aufgefaßt werden.

A4) Unendlich lange, ebene Stabwirbelreihen, gestaffelt (auf Luke) angeordnet auf zwei Ebenen im Abstande d, mit jeweils gegenläufigem Drehsinn.

Anordnung und Bezeichnungen siehe Bild 17 (nach Art der Kármánschen Wirbelstraße).
Über genaue Formeln für ebene Wirbelreihen siehe [6].

Das Bild 17 zeigt die Stromlinien der Absolutströmung. Es gibt 3 Bereiche: Zwei äußere, in denen die Flüssigkeitsteilchen um die Wirbelkerne kreisen und die keinen Beitrag zum Gesamtimpuls des Feldes liefern, sowie einen inneren Bereich, in dem eine Flüssigkeitsmenge Q (m^3/sek) nach rechts als Strahl transportiert wird. Im Unendlichen ($y \to \pm\infty$) ist die Strömung in Ruhe. Die drei Bereiche werden durch zwei im Bilde 17 gezeichnete Grenzstromflächen getrennt, die gerade jeweils zwischen zwei Wirbeln ins Unendliche gehen und sich dort schließen.

Dieses Bild ist einem Feld gleichmäßig verteilter Wirbelschichten (nach A2) gleichwertig, wenn die Strahlebenen annähernd denselben Abstand haben und die äquivalente Belegung $\gamma \simeq \Gamma/l$ gemacht wird.

Zur genauen Bestimmung von γ soll die (Impulsgleichheit voraussetzende) Vorschrift dienen, daß der Wirbelfluß $\left(\frac{d\Gamma}{dt}\right)_m$ erhalten bleiben soll.

Es ist genau:
Gruppengeschwindigkeit der Wirbelreihe

$$w_m = \frac{\Gamma}{2l}\cdot \mathrm{Tg}\,\frac{\pi d}{l}\;(\approx \gamma/2) \quad \ldots\ldots\ldots \quad (18)$$

Die Periode für einen Wirbeldurchgang durch eine (absolut) ruhende Ebene beträgt

$$T = \frac{l}{w_m} = \frac{2l^2}{\Gamma}\,\mathrm{Ctg}\,\frac{\pi d}{l}$$

damit gemäß Gl. 2 in Abschn. 2

$$\left(\frac{d\Gamma}{dt}\right)_m = \frac{\Gamma}{T} = \frac{\Gamma}{l}\,w_m = \frac{\Gamma^2}{2l^2}\,\mathrm{Tg}\,\frac{\pi d}{l} = \frac{\gamma^2}{2} \quad \text{(nach Gl. 16)}$$

dies dient zur Bestimmung von γ

$$j = \frac{\Gamma}{l}\cdot\sqrt{\mathrm{Tg}\,\frac{\pi d}{l}} \quad \ldots\ldots\ldots \quad (18a)$$

$\frac{j}{\Gamma/l} = \sqrt{\mathrm{Tg}\,\frac{\pi d}{l}}$ ist in Bild 18, untere Kurve dargestellt, für technisch in Frage kommende Bereiche von $d/l > 0{,}5$ ist die Abweichung von 1 praktisch vernachlässigbar.

Die im (dritten) Strahlbereich fließende Wassermasse errechnet sich aus [6]

$$\varrho \cdot Q_{abs} = \varrho\,\frac{\Gamma}{l}\cdot d = \varrho \cdot d\,\gamma\,\sqrt{\mathrm{Ctg}\,\frac{\pi d}{l}}$$

Diese Masse ist etwas $\left(\text{um } \sqrt{\mathrm{Ctg}\,\frac{\pi d}{l}}\right)$ größer als die Strahlmasse der äquivalenten Wirbelschicht γ. Beider Wirbelfluß ist aber identisch.

Um auch die Wassermassen im Strahl genau gleichzumachen, müßte man einen (kleineren) Abstand $d' = d\cdot\sqrt{\mathrm{Tg}\,\frac{\pi d}{l}}$ der Wirbelschichtebenen γ einführen. Der kinetische Austrittsverlust ist aber bei der Einzelwirbelanordnung stets größer.

A5) Unendlich lange, ebene Stabwirbelreihen, in Reihe (auf gleicher Höhe x) angeordnet auf 2 Ebenen im Abstande d, mit jeweils gegenläufigem Drehsinn.

Anordnung und Bezeichnungen siehe Bild 19.

Hierzu gelten qualitativ dieselben Bemerkungen, wie soeben in A4 gemacht.

Die äquivalente Flächenbelegung mit gleichförmigen Wirbelschichten γ ist wieder angenähert $\gamma \simeq \Gamma/l$.

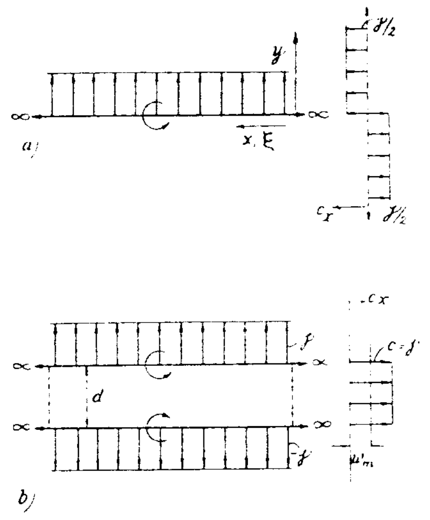

Abb. 16 Ebene Wirbelschichten mit ihrem Geschwindigkeitsfeld a) Einzelwirbelschichten b) Zwei parallele gegenläufige Schichten

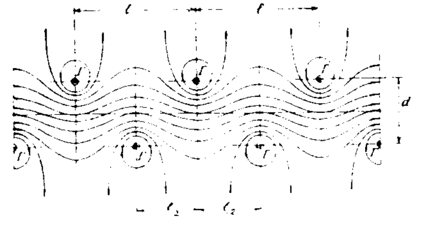

Abb. 17. Zwei gegenläufige parallele Reihen gerader Stabwirbel in gestaffelter Anordnung und ihr Geschwindigkeitsfeld. Gezeichnete Linien sind Stromlinien.

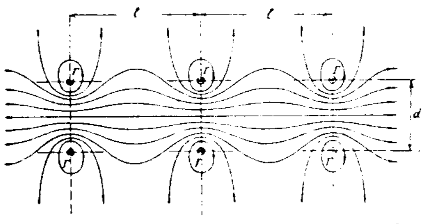

Abb. 19. Wie Abb. 17, die Wirbel sind in Reihe angeordnet.

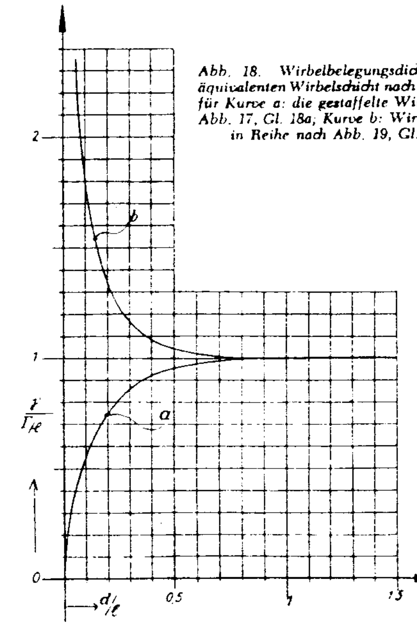

Abb. 18. Wirbelbelegungsdichte einer äquivalenten Wirbelschicht nach Abb. 16 b für Kurve a: die gestaffelte Wirbelreihe, Abb. 17, Gl. 18a; Kurve b: Wirbelreihen in Reihe nach Abb. 19, Gl. ???

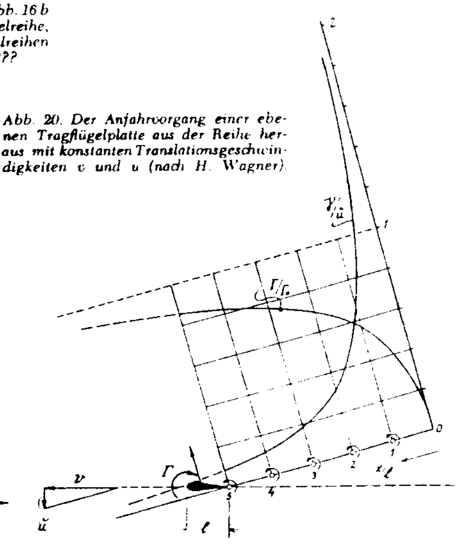

Abb. 20. Der Anfahrvorgang einer ebenen Tragflügelplatte aus der Reihe heraus mit konstanten Translationsgeschwindigkeiten v und u (nach H. Wagner).

Die genaue absolute Gruppengeschwindigkeit ist jetzt abweichend

$$w_m = \frac{\Gamma}{2l} \cdot \mathrm{Ctg}\,\frac{\pi d}{l} \sim \gamma/2 \quad \ldots\ldots\ldots (19)$$

Der mittlere Wirbelfluß ist entsprechend

$$\left(\frac{d\Gamma}{dt}\right)_m = \frac{\Gamma}{T} \cdot \frac{\Gamma^2}{2l^2}\,\mathrm{Ctg}\,\frac{\pi d}{l} = \frac{\gamma^2}{2} \quad \text{(nach Gl. 16)}$$

dies dient wieder zur Bestimmung von γ

$$\gamma = \frac{\Gamma}{l} \Big| \mathrm{Ctg}\,\frac{\pi d}{l} \quad \ldots\ldots\ldots (19a)$$

$\frac{\gamma}{\Gamma/l} = \Big| \mathrm{Ctg}\,\frac{\pi d}{l}$ ist aus Bild 18, obere Kurve zu ersehen. Die Abweichungen von 1 sind wieder praktisch vernachlässigbar für $\frac{d}{l} > 0{,}5$. Die Wassermasse im Strahl ist diesmal

$$\varrho \cdot Q_{abs} = \varrho \cdot \frac{\Gamma}{l}\,d = \varrho \cdot d \cdot \gamma \sqrt{\mathrm{Tg}\,\frac{\pi d}{l}}$$

etwas (um $\sqrt{\mathrm{Tg}\,\frac{\pi d}{l}}$) kleiner als die Masse der äquivalenten Wirbelschicht γ. Für nicht zu weitstehende Wirbel kann daher auch hier die Näherung $\gamma \cong \Gamma/l$ voll befriedigen.

A6) Wirbelringe auf Kreiszylindermantel des Durchmessers d. Hierfür kann (qualitativ als Näherung) Bild 19 als koaxialer Schnitt aufgefaßt werden. Je enger die Wirbel stehen ($l \leq d$), desto besser ist die Annäherung. Man wird deshalb auch hier die koaxiale Wirbelringreihe (gut) annähern können durch eine gleichmäßig mit $\gamma \cong \Gamma/l$ belegte, kreiszylindrische Wirbelschicht. Das Gleiche gilt für eckige oder sonst beliebig geformte, zylyndrische Strahlflächen. Es gelten dann sinngemäß alle Formeln von Abschn. A 5.

Genaue Formeln für Wirbelringe und Wirbelring-Belegungen finden sich in [7]. Es ist bemerkenswert, daß theoretisch die „Gruppengeschwindigkeit" w_m eines einzelnen Rin- -t klein werdendem Kernradius beliebig groß verden kann (nach Kelvin), daß sie aber sof e Werte annimmt, wie in [7] nachgewiesen wenn dieselbe Wirbelstärke auf eine endlich lange Mantelfläche als Wirbelbelegung verteilt wird. Letzteres Bild kommt den in dieser Studie auftretenden natürlichen Strömungen näher. Die endliche Gruppengeschwindigkeit von Wirbelringschichten w_m nähert sich den in Abschnitt A5 angegebenen Näherungswerten

$$w_m \cong \gamma/2 = \Gamma/2l$$

gut und stimmt im Grenzfall der gleichmäßigen Belegung γ genau damit überein.

B Wirbelschleppe anfahrender Tragflügel.

H. Wagner [8] hat das instationäre Arbeiten von ebenen Tragflügeln mit zeitlich veränderlichen Geschwindigkeiten und Anstellwinkeln untersucht.

Bild 20 zeigt das Verhalten eines am Orte x = O ruhenden und dann plötzlich mit der gleichmäßigen Geschwindigkeit v nach links und der Geschwindigkeit u nach unten bewegten (ebenen) Tragflügels der Länge l. Die hintere Abreißkante legt den Weg x nach schräg links unten zurück. Aus der (hinreichenden) Bedingung einer glatten Umströmung der Hinterkante errechnet sich die Entstehung von kantenparallelen, freien Wirbeln, welche als Kielwasser zurückbleiben und die ebene Fläche vom Beginn der Bewegung bis gerade zum jeweiligen Ort der Hinterkante gleichförmig abnehmend mit der gezeichneten, links drehenden Wirbelbelegung γ (m/sek) erfüllen. Ihre jeweilige Gesamtintensität $\Gamma = \int_0^x \gamma\, dx$ ist in einer Kurve gezeichnet. Um das Profil selbst besteht eine Zirkulation jeweils genau gleicher Größe Γ, aber umgekehrter, d. h. rechtsdrehender Richtung, in Übereinstimmung mit Helmholtz'Wirbelsätzen.

Hier interessiert nun, daß die Gesamtzirkulation Γ nach längerem Weg einem konstanten Werte Γ_0 zustrebt, der Zirkulation des stationär bewegten Flügels, daß aber dieser Wert keineswegs sofort nach Bewegungsbeginn erreicht wird. Gleichwertig hiermit ist aber die Feststellung, daß der wohl von L. Prandtl erstmalig untersuchte „Anfahrwirbel" keineswegs als Einzelwirbel nur am Ort x = O entsteht, sondern mit der gezeichneten Belegungskurve γ über den Abstrom verteilt ist, allerdings mit starker Konzentration nahe x = O. Diese Zusammenballung der Gesamtwirbelstärke Γ_0 nahe dem Anfahrpunkt wird desto stärker, je größer der Anfahrimpuls, also der Geschwindigkeitsbeginn, gemacht wird.

C Reine Potentialströmung der ebenen Borda'schen Mündung.

Die genaue potentialtheoretische Lösung findet sich in [9]. Das entstehende Strömungsbild ist in Bild 21 zu sehen. Da Potentialströmungen umkehrbar sind, könnte die Flüssigkeit aus dem Rohr austreten oder in ihm verschwinden. Wir betrachten letzteren Fall. Weit links im „Rohr" ist die Geschwindigkeit gleichförmig = c_k. Näher der Mündung zu sind die Geschwindigkeiten w auf der Symmetrieebene kleiner als c_k, während sie dort am Innenrand des Rohres größer als c_k werden ($w_i > c_k$), um schließlich direkt an der Kante der Mündung unendlich zu werden. Direkt auf der Außenseite des Rohres strömt die Flüssigkeit in umgekehrter Richtung, d. h. nach rechts auf die Mündung mit zunehmender Geschwindigkeit w_a, um gleichfalls an der Kante unendlich groß zu werden und sie dann ebenso schnell zu umströmen. Die Differenz der Werte $w_i - w_a = \gamma$ (Summe der Absolutwerte) sowie ihr arithmetisches Mittel $w_m = \frac{w_i + w_a}{2}$ sind im Bild 21 unten aufgetragen. In größerer Entfernung von der Mündung folgt die Außenströmung dem Gesetz einer punktförmigen Senke der Schluckfähigkeit $F \cdot c_k$.

Für die Anwendung in Abschnitt 4 empfiehlt es sich, dieses aus der Funktionentheorie abgelesene Strömungsbild umzudeuten.

Wir wollen annehmen, daß die als parallele Wände vorhandenen Begrenzungen der Mündung fortfallen. Wir setzen genau an ihre Stelle in einen überall (auch innerhalb des Rohres) ausgedehnten, ruhenden Flüssigkeitsraum zwei Wirbelschichten, der in Bild 21a gezeichneten Drehrichtung mit der genau gleich γ nach Bild 21b zunehmenden Wirbelbelegungschichte. Die Achse der einzelnen Stabwirbel $d\Gamma = \gamma\, dx$ steht senkrecht auf der Zeichenebene. Diese beiden Wirbelschichten werden nach Art von „festen" Stabwirbeln an ihrem Ort festgehalten gedacht. (Singularitätenprinzip bei Umströmung fester Körper). Sie induzieren nun ein Strömugsfeld, das genau dem einer idealen, rotationsfreien Borda-Mündung entspricht; die Wirbelverteilung γ ist (was auch direkt und eindeutig aus der Lösung einer Integralgleichung hätte erhalten werden können) gerade so angeordnet, daß überall auf den beiden Wirbelschichten die von den Wirbeln erzeugten Quergeschwidigkeiten w_y null werden, daß also dort die Grenzbedingung wie bei einer festen Wand erzielt wird. Mit zunehmender Entfernung von der Mündung nach links erhalten wir das schon im Anhang A2 erörterte Bild mit $\gamma = c_k$. Nahe der Mündung muß jedoch γ zur Einhaltung der Grenzbedingung ($w_y = O$) sehr viel größere Werte annehmen, um an der Kante sogar unendlich zu werden. Die Gesamtwirbelstärke Γ der Belegung von der Mündung bis zu einem Punkte x

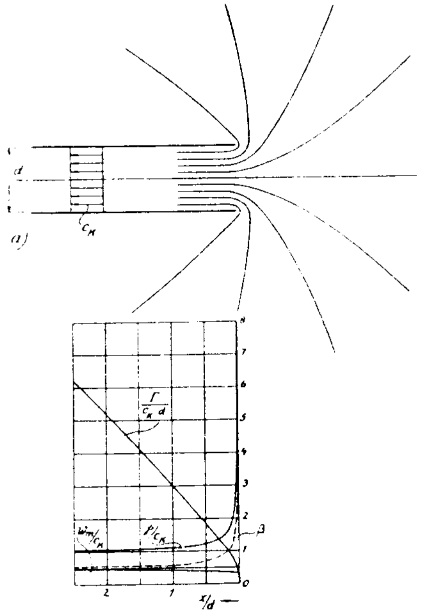

Abb. 21. Ebene Potentialströmung in und aus einer („Borda"-)Mündung. Oben: Stromlinienbild. Unten: Werte für Wirbellegung und mittlere Geschwindigkeit auf den ebenen Mündungswänden für zwei den Wänden äquivalente ebene Wirbelschichten.

ist jedoch endlich, als $\int_0^x \gamma\, dx$ errechnet und gleichfalls im Bild 21b aufgetragen. Die in Abschnitt 4 benötigte, in Bild 21b aufgetragene mittlere Geschwindigkeit w_m ist am Orte der oberen Wirbelschicht durch die untere induziert und umgekehrt. Sie bedeutet physikalisch die (bei der gezeichneten Drehrichtung nach links gerichtete) Eigengeschwindigkeit des Wirbelelementes γdx, sofern es sich nicht um (gedachte) feste Stabwirbel, sondern um (wirkliche) freie Wirbel handeln würde. Der Wert $\varrho \cdot \gamma \cdot w_m$ ergibt nach dem Kutta-Joukowskischen Auftriebssatz beiläufig den örtlichen Unterdruck innen an der Rohrwand. (Über den Mittelwertsfaktor β siehe Gl. 12a).

D Schwingung von Wassersäulen

D 1 Einfaches U-Rohr
Lösung siehe [10]

$$\omega = \frac{2\pi}{T} = \sqrt{\frac{2g}{l}}$$

D 2 Rohr kommunizierend mit sehr großem Gefäß (Halbraum)
Lösungsansatz durch Bernouillische Gleichung für nichtstationäre Vorgänge:

$$\frac{p}{\varrho} + \frac{w^2}{2} + \int_0^s \frac{dw}{dt} \cdot ds = \text{const}$$

Setzt man Reibungsfreiheit, d. h. gleiche Strömungsbilder bei Hin- und Herschwingen voraus und verwendet man für das örtliche Abnehmen der Geschwindigkeiten das Gesetz für Zuströmung zu einer Flächensenke $\pi R^2 w$, so ergibt sich die Eigenschwingungsfrequenz zu

$$\omega = \frac{2\pi}{T} = \sqrt{\frac{g}{l + R}}$$

Die Vergrößerung des Nenners l um R entspricht der „mitschwingenden Wassermasse" im großen Gefäß.

D 3 Oberflächenschwingung eines schmalen, endlich tiefenKanals, der eine freie Oberfläche hat und einseitig mit einem sehr großen Wasserraum verbunden ist.
Lösung grundsätzlich in [9, S. 409]
Bezeichnungen nach Bild 22c
Grundschwingungsfrequenz:

$$\omega = \frac{2\pi}{T} = \sqrt{\frac{g \cdot \pi}{2l} \cdot \mathrm{Tg}\,\frac{\pi \cdot h}{2l}}$$

Die stehende Welle hat einen festen Knotenpunkt an der rechten Mündung der Wände. Der Druck muß dort konstant bleiben, hingegen fließt das ganze, zwischen Wellental und -Berg unterzubringende Wasservolumen horizontal an der Mündung hin und her.

D 4 Rohr mit federnd gehaltenem Kolben als einseitigem Abschluß, nach rechts offen in Flüssigkeit:
Bezeichnung nach Bild 7b
Die Federkonstante sei τ (kg/m)

$$\omega = \frac{2\pi}{T} = \sqrt{\frac{\tau}{F(l + R) \cdot \varrho}}$$

F · R ist wieder die „mitschwingende Wassermenge".

D 5 Schwingende Säule des elastischen Wassers.
Siehe Bild 7c
Schallgeschwindigkeit

$$a = \sqrt{\frac{1}{\frac{\varrho}{\varepsilon} + \frac{\varrho}{E} \cdot \frac{D}{\delta}}} \text{ mit}$$

$\varepsilon = 2{,}07 \cdot 10^8\ \mathrm{kg/m^2}$ El.-Modul von Wasser
$E = 2{,}15 \cdot 10^{10}\ \mathrm{kg/m^2}$ El.-Modul von Stahl
für Wasser in Rohren des Durchmessers D und der Wandstärke δ
$500 < a < 1400$ m/sek

$$\text{Dann ist } \omega = \frac{2\pi}{T} = 2\pi n = \frac{\pi \cdot a}{l}$$

Von diesem Aufsatz werden nach Erscheinen des Heftes 500 Sonderdrucke hergestellt, von denen 200 durch Vermittlung des Instituts für Strömungslehre und Strömungsmaschinen der Technischen Hochschule Karlsruhe für Interessenten zum Bezuge freigestellt werden.

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PRELIMINARY CHARACTERIZATION OF ONE-AXIS ACOUSTIC SYSTEM

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PRELIMINARY CHARACTERIZATION OF A ONE-AXIS ACOUSTIC SYSTEM

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and H. W. Parker
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January 1979

NASA

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16. ABSTRACT

Measurements have been made of the acoustic fields and levitation forces produced along the axis of a single-axis resonance system. The system consisted of a St. Clair generator and a planar reflector. The levitation force was measured for bodies of various sizes and geometries (i.e., spheres, cylinders, and discs). The force was found to be roughly proportional to the volume of the body until the characteristic body radius reaches $\sim 2/k$ (k = wave number).

The acoustic pressures along the axis were modeled using Huygens principle and a method of imaging to approximate multiple reflections. The modeled pressures were found to be in reasonable agreement with those measured with a calibrated microphone.

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TECHNICAL MEMORANDUM 78213

PRELIMINARY CHARACTERIZATION OF A ONE-AXIS ACOUSTIC SYSTEM

INTRODUCTION

Many useful materials are adversely affected by contamination resulting from contact with container walls during their manufacturing. The microgravity environment that is available in an orbital facility such as Shuttle/Spacelab allows for the positioning of material away from container walls in a relatively simple fashion. One can then process materials in space and avoid many of the problems that limit operations on the ground. Active positioning of the material within the orbiting facility would be required to prevent it from drifting into contact with the walls and to maintain its position relative to any heat sources. Several positioning systems are being developed for use in the microgravity environment. These systems are based upon terrestrial levitation devices and for the present can be grouped into two categories: electromagnetic (EM) positioning systems [1] and acoustic positioning systems [2,3,4].

To completely characterize the material processed in space, the operation of the positioning system and its influence on material properties (e.g., influence of stirring) have to be fully understood. In addition, the capabilities of terrestrial levitation devices have to be documented so that the rationale for conducting containerless experiments in space can be firmly established. Because of the wide usage of EM systems, their limitations appear to be well known, with the most severe being the extreme difficulty in separately controlling the heating of the specimen and the levitation force [5]. Acoustic systems have been used only in a limited fashion to process/manufacture materials on Earth, and little is known of their capabilities.

Various terrestrial levitation and microgravity positioning systems are being characterized at NASA's Marshall Space Flight Center. This report presents some of the initial results characterizing a one-axis acoustic resonance system. This system was chosen in part because it is the basis of an acoustic positioning/furnace facility which has stably positioned specimens at elevated temperatures ($\simeq$ 1600°C) during short-duration (10 sec) microgravity KC-135 parabolic flight trajectories. In addition, control and understanding of the acoustic fields and acoustic-generated forces are simplified by having only one sound source.

THEORY

The forces acting on a small body (i.e., $kR << 1$ where k is the wave number and R the body radius) in an acoustic field have been derived by several authors [6,7,8]. By time averaging the pressure over the surface, the acoustic radiation force on the body in a plane stationary wave was found to be

$$F_a = 4\pi\bar{E}kR^3 \sin(2kx)\ F(\rho, \rho_o, c, c_o) \tag{1}$$

where $\bar{E}$ = average acoustic energy density, ρ_o = body density, ρ = density of medium, c_o = sound speed in body, c = sound speed in air, and $F(\rho, \rho_o, c, c_o) \sim 5/6$ for a solid body in air to levitate a body in a stationary plane wave against the gravitational force

$$F_a \gtrsim \frac{4}{3}\pi R^3 \rho_o g$$

where g is the gravitational acceleration. The preceding equation can be rearranged to yield

$$\frac{\bar{E}k}{\rho_o g} \sin(2kx) \gtrsim \frac{2}{5} \quad .$$

Seeking the position in the field where the maximum levitation occurs, we have

$$k\,\bar{E} \gtrsim 0.4\,\rho_o g \quad .$$

It is of interest to calculate the acoustic field necessary to levitate a body to help establish the parameters of the one-axis system (e.g., intensity of the sound source). We are interested in studying the fields and forces along the axis because previous results indicated that stable levitation nodes could be found on the axis [4]. The acoustic fields along the axis of the one-axis resonance device should be approximated by a stationary plane wave and, therefore, the preceding derivations should be applicable.

Many materials investigations desire relatively large samples (~ 1 cm dimensions) to be able to measure bulk properties. Therefore, the wave number k of the system should be $\sim 3\ \text{cm}^{-1}$. (At diameters $\gtrsim \lambda/2$, the body begins to overlap successive levitation nodes in the standing wave pattern, resulting in a reduction in forces on the body.) Assuming a characteristic body density of $\sim 10\ \text{gm/cm}^3$, $\bar{E}$ is determined to be $\sim 100\ \text{J/m}^3$. In air at STP, this would result in the acoustic field having an intensity of ~ 165 dB. Higher intensity is needed at the higher temperatures to compensate for the increase in sound speed. Even with the amplification system inherent with a system at resonance, a high-intensity sound source is required. The system discussed in this report is a St. Clair generator which can produce a high-intensity field in a gas [9] and can be easily used in a furnace configuration.

The acoustic standing wave pattern in an open one-axis system should be different from that expected in an enclosure because of the leakage of energy from the sides. To model the fields (and hence the forces) expected in these unique systems, the pressure wave generated by the driver was assumed to be reflected from the reflector/driver several times. The contributions to the pressure at a field point (between the driver and reflector) due to successive reflections were approximated by images of the driver at successively greater distances. This is depicted in Figure 1, where Huygens principle of superposition is used to calculate the total pressure at a field point, $\vec{r}$; i.e.,

$$P_T = \text{Pressure total}\ (\vec{r})\ \alpha \int\limits_{\text{surface of driver}} \frac{e^{i\vec{k}\cdot(\vec{r}-\vec{r}\,')}}{|\vec{r}-\vec{r}\,'|}\, d\vec{r}\,'$$

$$+ \sum_{i=1}^{10} \int\limits_{\text{surface of images } i} \frac{e^{i\vec{k}\cdot(\vec{r}-\vec{r}_i\,')}}{|\vec{r}-\vec{r}_i\,'|}\, d\vec{r}_i\,'$$

where $\vec{r}\,'$, $\vec{r}_i\,'$ are points on the surface of the driver and its images. For the problem of interest the number of reflections was limited to 10. This number of reflections appeared to approximate the completely calculated pressure to within ±15 percent.

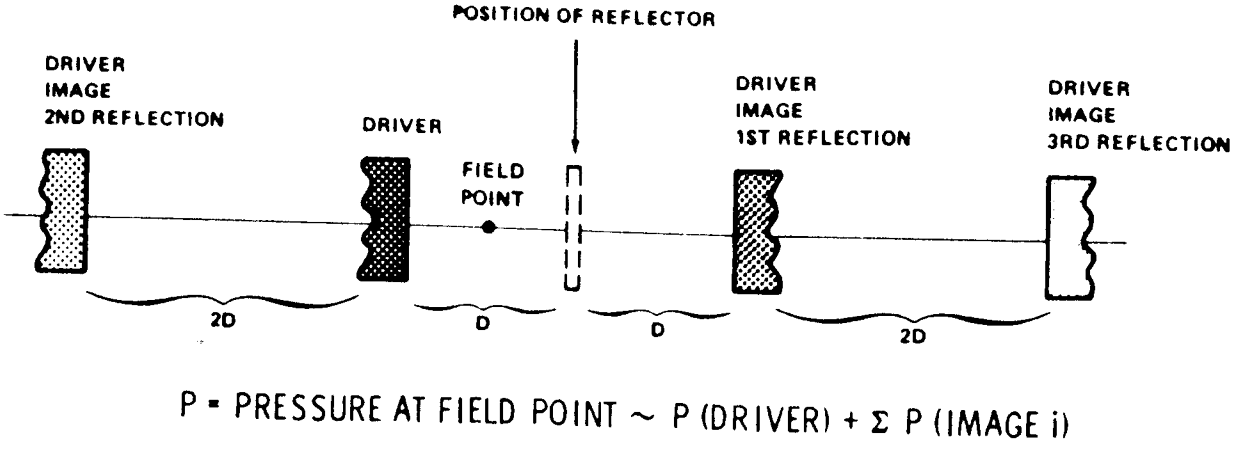

Figure 1. Conceptual schematic depicting the method of calculating the pressure on the axis between the driver and reflector. (The initial pressure pulse produced by the driver is assured to bounce between the reflector/driver several times. The effect of the bounces is approximated by a series of driver images on either side of the field point. Seven images and the driver were used to calculate the pressure.)

APPARATUS

The one-axis acoustic system consists of a sound source and a plane reflecting surface, as shown in Figure 2. The sound source is a St. Clair generator, which consists of a solid aluminum cylinder supported at its mid-section by a flexible aluminum diaphragm and driven at the resonance frequency of the aluminum bar by electromagnetic coupling [10]. This particular device, supplied by Intersonics, Incorporated,[1] has a resonance frequency of ~15 kHz, resulting in a 2.4-cm wavelength in air.

A hollow tube was used to suspend the reflector above and parallel to the piston face. The driver-reflector separation was easily adjustable; the driver and reflector diameters are ≈ 3.5 cm. A small bead was suspended along the axis of the acoustic field on a thin wire extending through a small hole in the reflector and up the tube to the arm of a Cahn balance.[2] This balance operates

---

[1] Intersonics, Inc., 490 E. Ohio, Chicago, Illinois 60611.

[2] Cahn Instrument Company, 7500 Jefferson Street, Paramount, California 90723.

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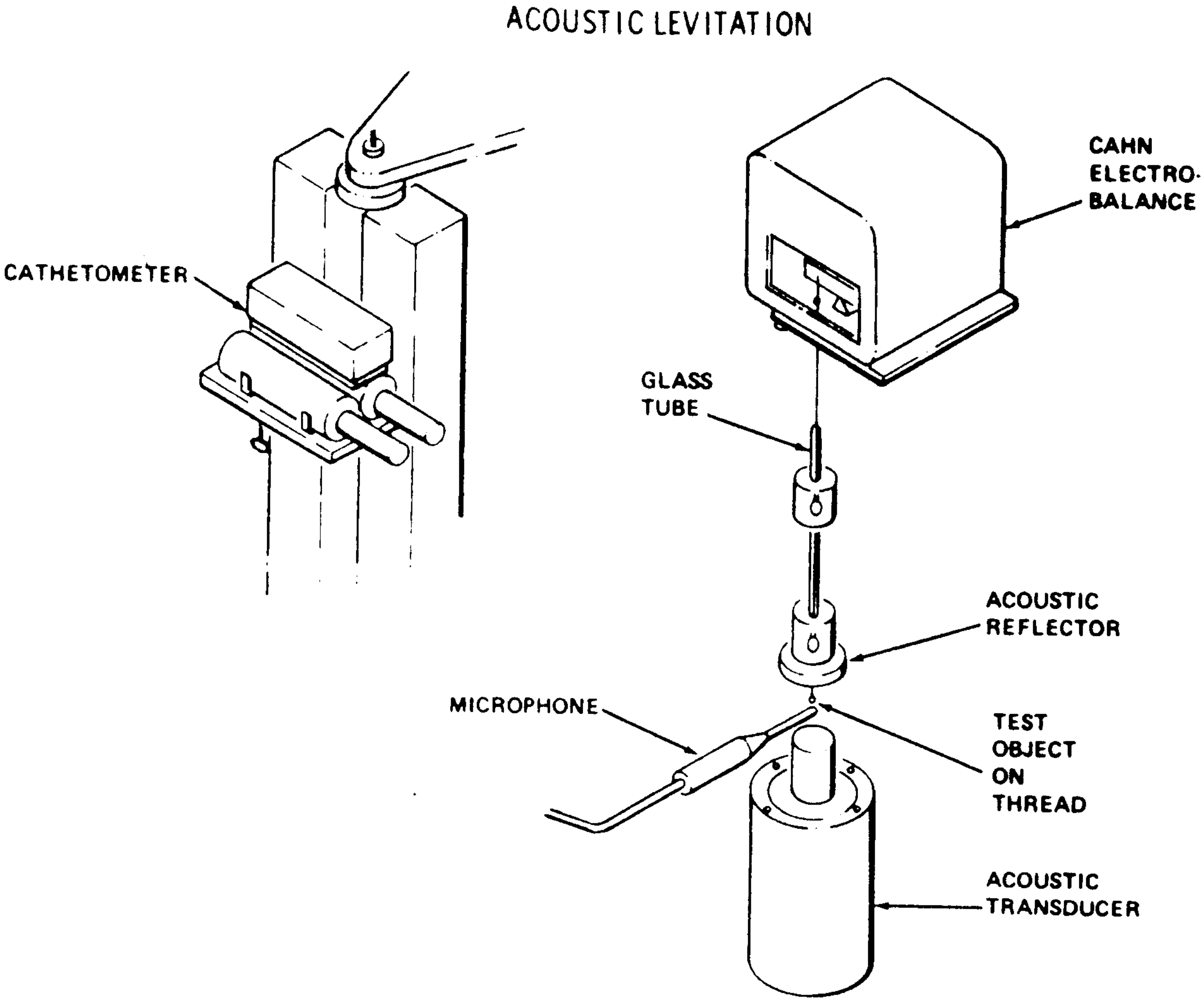

Figure 2. Schematic of the one-axis acoustic system and measuring instrumentation.

on the null-balance principle. Deflection of the balance beam resulting from force changes is opposed by a dc torque motor which quickly restores the beam to its original balance position. Thus, the force is always measured at the desired point in the device.

The acoustic field was measured using a 0.64-cm (0.25-in.) diameter Bruel and Kjaer condenser microphone and readout. The vertical positions of the vibrator, reflector, and bead were measured with a cathetometer.

RESULTS AND DISCUSSION

Figure 3 is an example of the forces measured on a body positioned on the axis of the system. The body was a sphere 0.8 cm in diameter, and the radiated sound intensity of the driver with the reflector removed was ~0.1 W/cm^2. Figure 3 shows two examples of forces on the body: In one example the system was at resonance (i.e., driver/reflector separation = 5.0 cm), and in the other the separation was increased by 0.2 cm to 5.2 cm. The levitation force is proportional to the pressure gradient across the body. As the pressures generated by the vibrator are amplified the most with the system at resonance, the maximum levitation force on a body should occur with the system at resonance. The influence of the resonance mode of operation on the levitation force is clearly indicated in Figure 3. The maximum forces at resonance are ~4 times greater than when the separation is increased 0.2 cm.

Data similar to those of Figure 3 were taken for several body sizes and geometries; all other conditions were identical for the system at resonance. Figure 4 shows the forces on these bodies at the position of maximum levitation force (i.e., ~1.6 cm). The force is plotted versus body volume with the geometry of the bodies indicated in the figure. These data show that for a range of shapes and sizes the levitation force is roughly proportional to body volume. This relationship holds until the characteristic body diameter reaches ~(1/4 - 1/2)λ.

At diameters $>\lambda/2$, the body would begin to overlap successive levitation nodes, resulting in a reduction of the forces on the body. This can be seen in Figure 4, where the force on the 1.2-cm diameter sphere is significantly less than could be expected based on the data for the smaller bodies. The force on a highly irregular body depends to some extent on its orientation in the acoustic pressure field. Figure 4 shows the forces on a 0.5 cm diameter by 0.12 cm thick disc. The lifting force on this disc when oriented with its axis perpendicular to the driver face is twice that experienced when it is oriented with its axis parallel to the driver face. This is in agreement with previous results which indicated that a large levitation force could be obtained on a thin disc whose axis was perpendicular to the driver force [4].

Based on the data of Figure 4, it appears that for bodies of interest to materials scientists (e.g., spheres, right cylinders) the levitation force can be roughly calculated from the body volume. The dependence of this force on the body volume is in agreement with the derivation discussed in theory.

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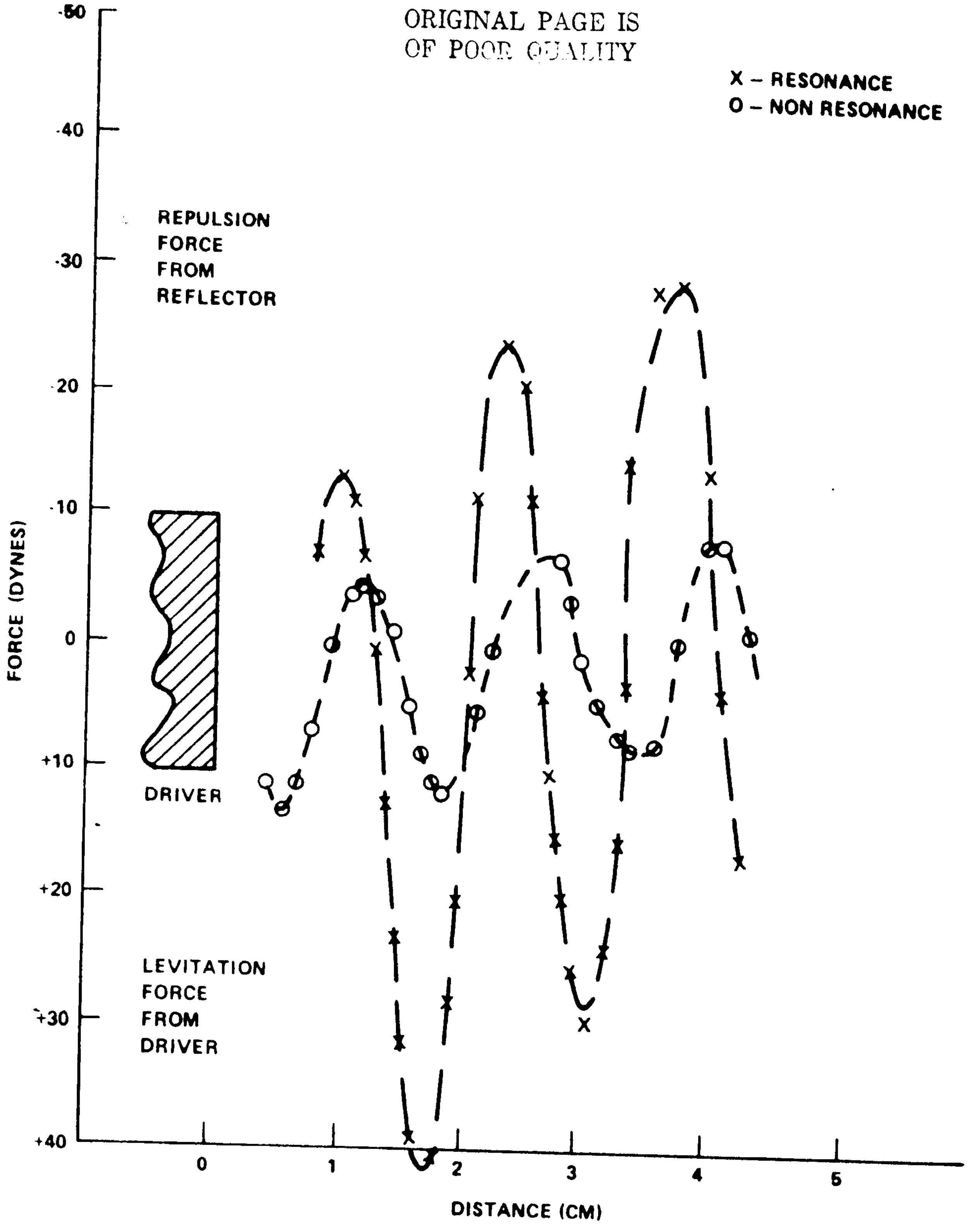

Figure 3. Forces measured on an 0.8-cm diameter sphere along the axis versus distance from the driver. (At resonance the reflector was ~5.0 cm from the driver; for nonresonance, the reflector was ~5.2 cm from the driver.)

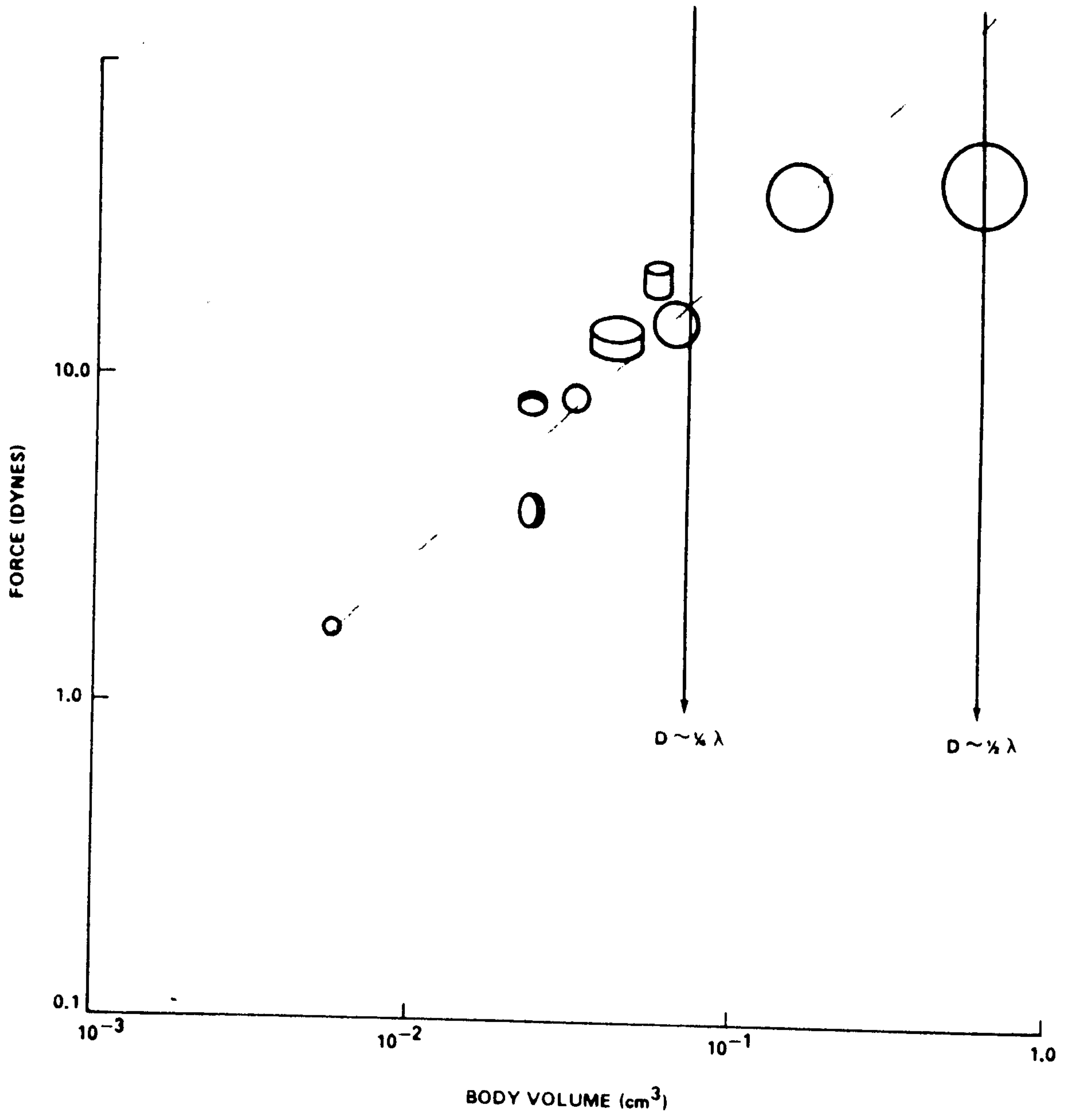

Figure 4. Axial force on body versus body volume. [The system is at resonance (reflector ~5 cm from driver) and the force was measured ~1.6 cm from driver. The body geometry is shown in the figure. The volume of spheres whose diameter (D) equals 1/4 and 1/2 λ (λ = 2.4 cm) is shown. The dotted line is hand drawn to show the "best fit" of the data for D < 1/2 λ.]

It should be noted that the theoretical expressions were derived for $kR << 1$. Figure 4 indicates that under certain conditions, the proportionality holds until $R \approx \lambda/3$ or $kR \approx 2$.

Figure 5 shows the pattern of rms sound pressures (p) measured along the axis for conditions similar to that of Figure 3 (resonance). The pressure is normalized to the free piston pressure measured ~1 cm above the driver ($\sim 0.64 \times 10^4$ dyne/cm^2). The pattern is similar to that expected in a "plane wave" resonance device. A comparison of Figures 3 and 5 shows that the zero force point for stable levitation occurs at the pressure nodes as predicted by theory. It was of interest to compare the measured forces with that predicted from the theoretical treatments of the acoustic forces in a stationary plane wave. The body used in this instance was the 0.42-cm diameter sphere which, from the data of Figure 4, experiences a force of ~9.5 dynes at ~1.6 cm from the driver. Taking the expression for the maximum acoustic-generated forces from equation (1) [i.e., $\sin(2kx) = 1$] and using the appropriate acoustic field parameters (i.e., $p = 10^4$ dyne/cm^2 and $\lambda = 2.4$ cm), the force is calculated at ≈ 25 dynes. This force was also calculated from the axial gradient of the acoustic radiation energy [i.e., $F \alpha \; d/dx(\bar{p}^2 - \rho^2 c^2 \bar{V}^2)$], where $\bar{V}$ is the rms particle velocity. Assuming that $V = -1/i\omega\rho \; dp/dx$, the force can be calculated from the experimental data of Figure 5 (e.g., dp/dx and p at 1.6 cm). These calculations yield a force of 8 dynes.

The assumption that a plane standing wave can describe the acoustic fields along the device should be considered only an approximation. The force calculated from the gradient of the acoustic radiation energy is in closer agreement with the measurement than the force calculated from equation (1), which is derived for a plane wave system. The relative discrepancies between the calculated values of the forces would indicate the limitations of using a plane wave approximation.

Figure 4 also shows a computer calculation of the rms pressures produced along the axis of the one-axis system. The pressure wave generated by the driver was assumed to reflect off the reflector/driver several times, and Huygens principle of superposition was used to calculate the pressure at a point (Fig. 1). The theoretical pressures were normalized to the free driver pressure modeled at 1 cm on the axis above the face. There is general agreement between the measured and modeled pressures shown in Figure 5, giving some degree of confidence in using this approach to predict the acoustic fields under other conditions (e.g., off-axis and at high temperatures when the piston diameter $< \lambda$).

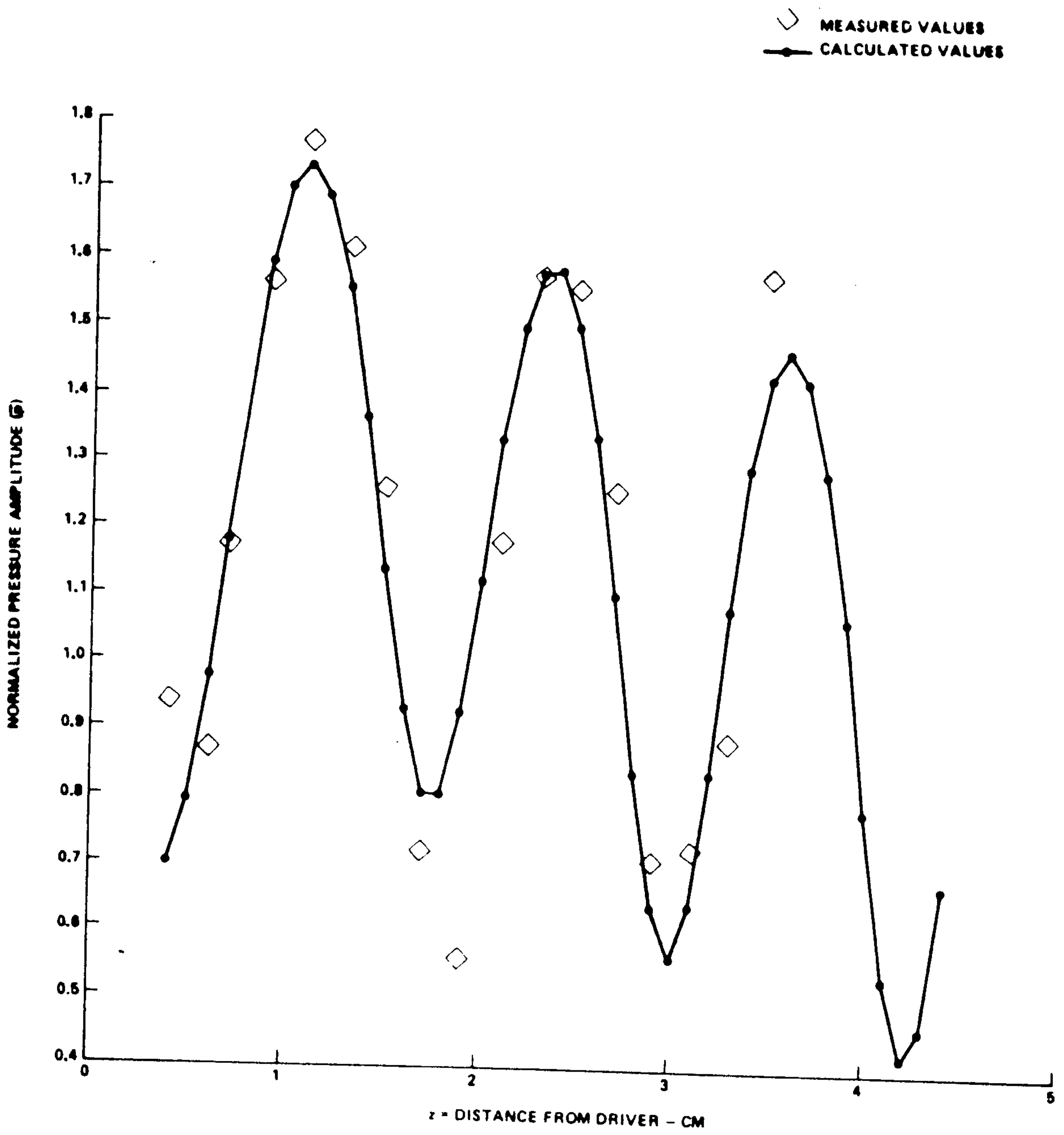

Figure 5. rms pressure along the axis of the device for the system at resonance (driver/reflector distance ~5 cm). [The experimental data were normalized to the pressure produced by the free-standing driver 1 cm from its face (~0.64 × 10^4 dyne/cm^2). The calculated pressures were normalized independently to that of a free-standing driver theoretically modeled 1 cm from its face.]

In the future the computer code of the pressure will be expanded to completely describe the acoustic field in the device (i.e., the axial and radial variations). Also, the acoustic fields and forces will be experimentally mapped and compared with the modeled fields to better understand phenomena such as: (1) the shape and strength of the levitation points, (2) the influence of drivers and reflectors of different sizes, and (3) the limitation of such devices at high temperatures.

SUMMARY

The acoustic fields and acoustically generated forces were characterized in a one-axis resonance acoustic system consisting of a flat piston-type sound source and a planar reflector. These measurements were made as part of a program to determine the capabilities of various ground-based levitation systems.

Measurements of the acoustic force were made on bodies of various sizes and geometries suspended from the arm of a sensitive electrobalance. The force was determined to be roughly proportional to body volume until the characteristic body diameter reached $\sim(1/4 - 1/2)\lambda$. This indicated that the body density is an important parameter in determining whether a material can be processed in an acoustic levitator on the ground.

A simple one-dimensional model of the acoustic field (based on Huygens principle and taking into account multiple reflections between the driver and reflector) was used to calculate the pressure pattern on the axis of the device. These axial pressure variations were also measured and found to be in agreement with the theoretically modeled values of pressure variation.

One of the objectives of the present program is to determine the capabilities of terrestrial acoustic processing systems, and future work will investigate (theoretically and experimentally) the acoustic fields and forces on and off axis to characterize a one-axis device. This effort will also involve studying the operation of a one-axis system in an enclosed furnace at elevated temperatures as well as analyzing these devices at room temperature.

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APPROVAL

PRELIMINARY CHARACTERIZATION OF A ONE-AXIS ACOUSTIC SYSTEM

By W. A. Oran, D. A. Reiss, L. H. Berge,
and H. W. Parker

The information in this report has been reviewed for technical content. Review of any information concerning Department of Defense or nuclear energy activities or programs has been made by the MSFC Security Classification Officer. This report, in its entirety, has been determined to be unclassified.

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THE INFLUENCE OF WALL RESONANCES ON THE LEVITATION OF OBJECTS IN A SINGLE-AXIS ACOUSTIC PROCESSING CHAMBER

GEORGE C. MARSHALL SPACE FLIGHT CENTER
MARSHALL SPACE FLIGHT CENTER, AL

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By Bruce B. Ross
Space Sciences Laboratory

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16. ABSTRACT

Instabilities have apparently been observed in high-temperature, single-axis acoustic processing chambers. It is hypothesized that at certain temperatures strong wall resonances are generated within the processing chamber itself and that these transverse resonances are sufficient to disrupt the levitation "well." This investigation indicates that, if these wall resonances do exist in the processing chamber, they are apparently not strong enough to cause instabilities in the levitation well.

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INTRODUCTION

A sample of gallia/calcia/silica glass was lost from the levitation "well" of the single-axis acoustic system in the SPAR VI sounding rocket flight [1]. Observation of the flight film indicated that an instability may have developed in the levitation well. It has been hypothesized that the acoustic field may have induced a radial resonant condition sufficient to disrupt the levitation well when the processing chamber reached certain temperatures. A partial simulation of this phenomenon was performed by varying the dimensions of a processing chamber at room temperature in order to simulate the temperature variation effect on the wavelength of sound in the processing medium. At certain wall separations a resonance condition was established and the effect on the levitation well noted.

APPARATUS

The experimental apparatus is shown schematically in Figure 1. A St. Clair sound generator with a stepped driver of 3.5 cm head diameter is used as the acoustic source. A levitation node is formed by interfering the sound field with a 2.0 cm^2 reflector. Measurements taken with B&K sound pressure equipment included axial sound pressure distributions between driver and reflector for resonance and antiresonance conditions as shown in Figure 2 (a,b). The sound pressure was measured radially from the common levitation node to the wall of the processing chamber. The walls were positioned at three different separations corresponding to possible resonance conditions of 3, 4, and 5 times the wavelength of the axial sound pressure distribution (λ = 2.256 cm), as shown in Figures 3 (a,b,c) and 4 (a,b,c). The dimensions include the actual dimensions of the SPAR VI processing chamber (10.8 cm × 10.8 cm × 11.4 cm) [2]. Also, any fluctuations of the sound pressure level in the levitation node resulting in instabilities in a sample being levitated were noted as the wall separation of the processing chamber was constantly varied.

AXIAL SOUND FIELD

In the resonance condition the levitation node is one quarter of a wavelength below the reflector. This is verified by the data in Figure 2a. If the reflector is moved by one quarter of a wavelength to the antiresonance condition, the worst case of a typical off-resonance condition, the levitation node is also located one quarter of a wavelength

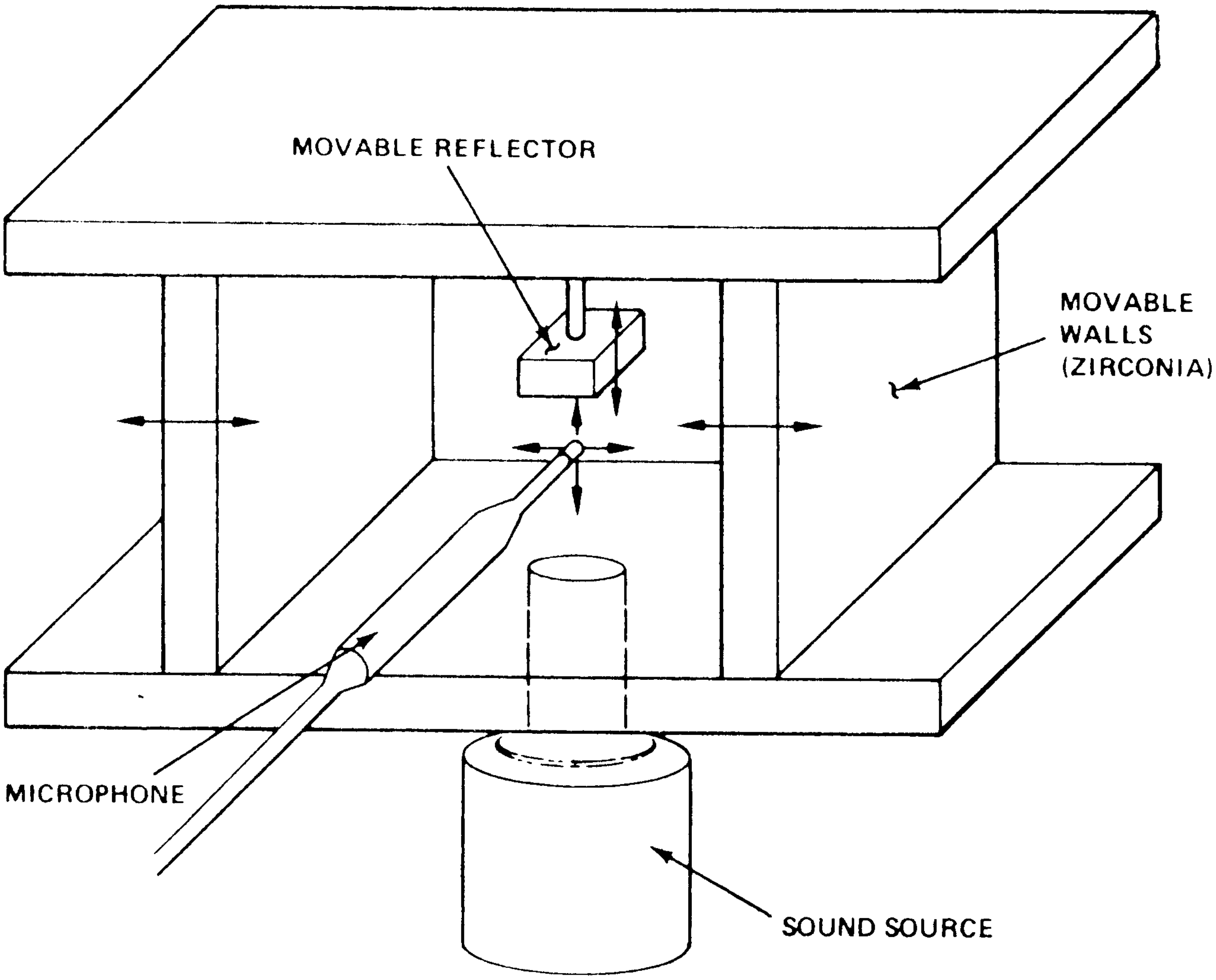
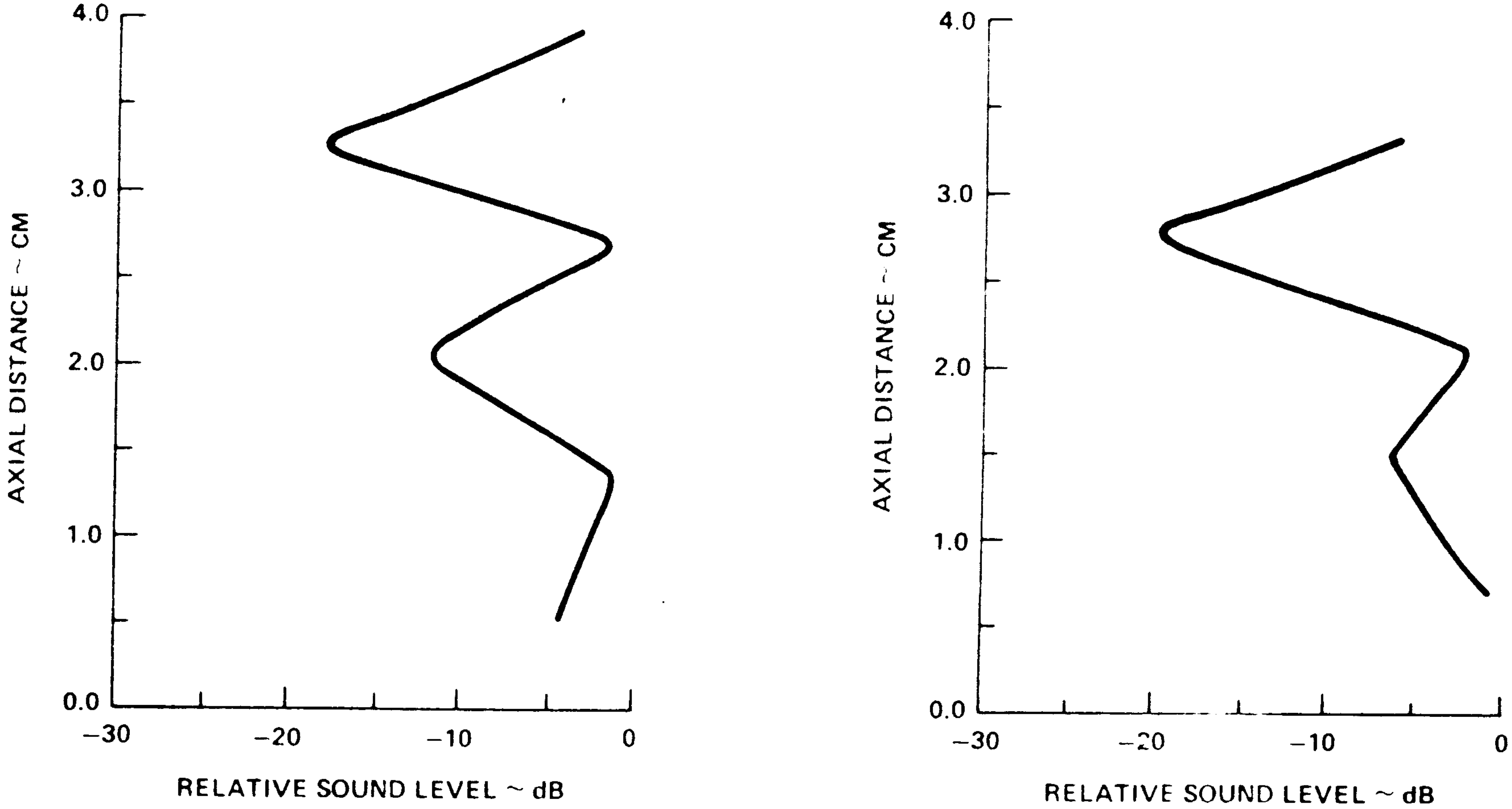


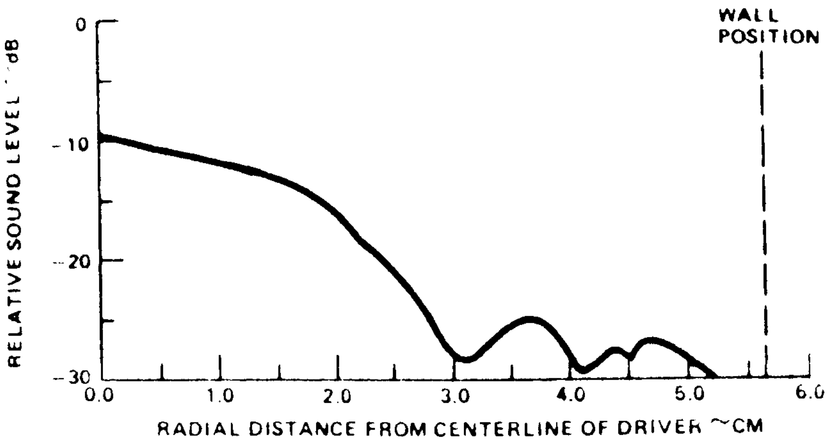
Figure 1. Simulated acoustic processing chamber.



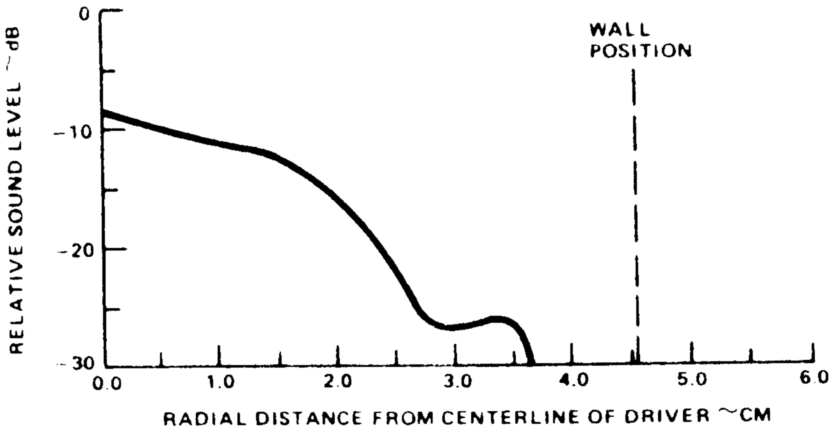
A. AXIAL DISTANCE ABOVE DRIVER VERSUS RELATIVE SOUND LEVEL FOR RESONANCE CASE

B. AXIAL DISTANCE ABOVE DRIVER VERSUS RELATIVE SOUND LEVEL FOR ANTIRESONANCE CASE

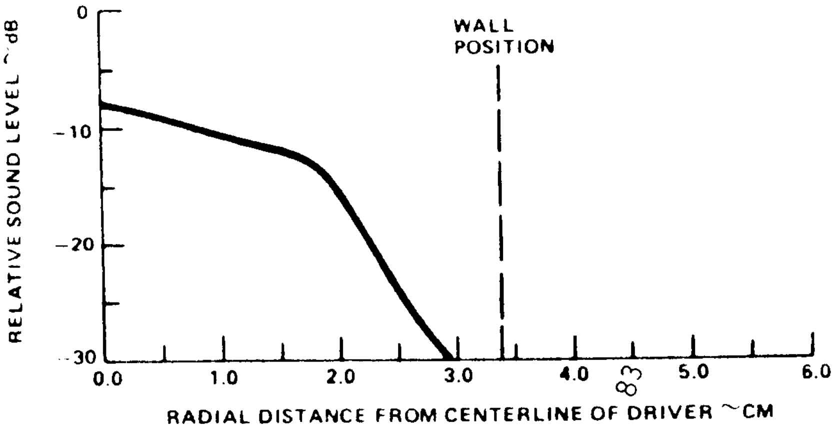
Figure 2. Axial distance versus relative sound level.



A RELATIVE SOUND LEVEL VERSUS RADIAL DISTANCE FOR A WALL SEPARATION OF 5λ IN THE ANTIRESONANCE CASE



B. RELATIVE SOUND LEVEL VERSUS RADIAL DISTANCE FOR A WALL SEPARATION OF 4 λ IN THE ANTIRESONANCE CASE



C RELATIVE SOUND LEVEL VERSUS RADIAL DISTANCE FOR A WALL SEPARATION OF 3 λ IN THE ANTIRESONANCE CASE

Figure 3. Relative sound level versus radial distance in antiresonance case.

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APPROVAL

THE INFLUENCE OF WALL RESONANCES ON THE LEVITATION OF OBJECTS IN A SINGLE-AXIS ACOUSTIC PROCESSING CHAMBER

By Bruce B. Ross

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KEELY: PIONEER OF GRAVITY CONTROL

Keely lived one hundred years too soon. If he were alive today he would be the foremost leader in the field of acoustical levitation. NASA has only now begun to realize the potential of acoustical levitation in minor utility roles. Working alone in a then totally uncharted science one hundred years ago, Keely is outstanding.

He was working in the acoustical frequency range of what is generically called gravity control. What we call levitation or antigravity is really the control of gravity itself. There are many frequency ranges in the EM spectrum which have gravity control frequencies in them, but probably the easiest gravity control frequency range for the interested amateur to use (which is what Keely used) is the lowest major gravity standing wave section between 7.8 Hz and 60,000 Hz. At the low end of this first major gravity standing wave section is the audio frequency band in which Keely worked. The general frequencies that are gravity control in the audio frequency bandare 7.8 Hz, 73.5 Hz, 690 Hz, and 6,450 Hz. The ones at 7.8 Hz and 690 Hz are the strongest, with the ones at 73.5 Hz and 6,450 Hz usable with more power. Any good audio oscillator can generate the 690 Hz and 6,450 Hz sine-wave signals. Causing levitation is then only a matter of multiple speakers or transducers to radiate the signal properly arranged. The term "properly arranged" signal sources is a bit complex. Usually an arrangement of 6-8 or more signal sources (speakers, etc.) equidistant from the object to be levitated is required for gravity control. These multiple sources are equally spaced around the levitation object in 90 degree or 180 degree arrangements. The three critical factors are the signal feed-in angles, the signals phasing, and the proper power levels. Now you can appreciate all the problems and failures that Keely faced in his day. How he ever got it to work at all is amazing! But then again, the Tibetans levitate large rocks with these audio frequency methods using only drums and trumpets. I would venture a guess that the Tibetans use 73.5 Hz and 690 Hz with those particular musical instruments. The musical score for the Tibetan levitation must sound very strange to achieve the proper phasing and power criteria.

In summation, Keely was right about the prospects for levitation but regrettably he was one hundred years too early. A great deal of his work can be recovered if proper understanding of his peculiar vocabulary can be achieved. He was a pioneer of the first rank in real science. I am sure that in the years to come he and his works will receive the honor that they so greatly deserve.

Permission is given to anyone to publish all or any part of the above data.

Note: I am enclosing a copy of the Tibetan sound levitation method. Please do not publish my address. I do not wish to answer questions from readers.

Richard LeFors Clark, Ph.D.

Richard LeFors Clark, Ph.D.
April 22, 1986

A high-powered siren for stable acoustic levitation of dense materials in the Earth's gravity

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Levitation of large dense samples (e.g., 1-cm diameter steel balls) has been performed in a 1-g environment. A siren was used to study the effects of reflector geometry and variable-frequency operation in order to attain stable acoustic positioning. The harmonic content and spatial distribution of the acoustic field have been investigated. The best stability was obtained with an open reflector system, using a flat lower reflector and a slightly concave upper reflector while operating at a frequency slightly below resonance.

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INTRODUCTION

Acoustic levitation is a potential tool for the containerless processing of materials.[1] Although great progress has been made with levitation and manipulation of samples in a microgravity environment and with levitation of low-density samples in the Earth's gravity,[2,3] levitation of large dense samples in the Earth's gravity still poses severe challenges as it requires both high power levels and stability.

Low-density objects (e.g., styrofoam) have been levitated against the Earth's gravity in gaseous medium.[2,3] The acoustic power was delivered to resonant chambers by conventional electroacoustic transducers (loudspeaker coils and piezoelectric devices). Although they provided good control of the acoustic signals, the power levels were limited to 140–150 dB, which was not adequate to levitate more dense objects (e.g., steel).

Recently, Lee and Feng[4] demonstrated the stable levitation of high-density submillimeter objects against the Earth's gravity using a focusing radiator. By concentrating the energy from 130 individual lead zirconate-titanate (PZT) transducers, they were able to achieve sound levels of 172 dB.

St. Clair[5] produced high intensities for the acoustic levitation of heavy objects by using a resonant cylinder. This device made use of the high-Q ($\sim 16\,000$) resonance of an aluminum cylinder and a movable reflector. Since it operated at a single frequency, the position of the reflector was adjusted to compensate for changes in the acoustic resonance due to temperature, humidity, and sample size.

An acoustic siren has the singular advantage of being able to produce a high-intensity acoustic field over a wide range of frequencies. Allen and Rudnick[6] demonstrated the levitation of high-density objects of a few centimeters in size by using a high-powered siren. A properly designed siren can produce extremely high intensities, with an efficiency of over 50%.[7] The siren is particularly efficient when its high acoustical impedance is matched to the impedance of the air by an exponential horn.[6,7]

Allen and Rudnick[6] reported that their system required constant siren frequency adjustments to levitate an object. Even with the careful adjustment of the siren frequency, frequently large oscillations of the sample developed for no apparent reason, and the object was eventually ejected from the field.

Containerless processing requires levitating objects for long periods of time without close operator attention. The goal of this study was to identify the causes of the instability and to remove those causes so that the siren could become a dependable acoustic levitation facility for the containerless processing of high-density objects larger than 1 mm in the Earth's gravity.

I. THE SIREN SYSTEM

A siren produces sound by periodically interrupting an otherwise steady flow of compressed air. The siren and levitation system used in these studies are shown in Fig. 1. The siren is similar to the one used by Allen and Rudnick.[6] The rotor and stator each have 100 holes of 2.54-mm diameter on a 76.4-mm radius, and the rotor is driven by a 1.12-kW (1.5-hp) variable-speed motor. The acoustic frequency is determined by the speed of the motor, which must be accurately controlled. The acoustic impedance of the siren is matched to that of the air by a tapered horn section designed to approximate the exponential horn of Allen and Rudnick.[7]

In its conventional implementation, the siren is a ring source which, when used with its axis of rotation vertical, gives a spatial pressure distribution that can be used to levitate objects. However, the high harmonic content of the acoustic pressure waveform excites multiple modes in the resonant reflector system, producing conflicting force distributions.

The temperature and humidity of the air were stabilized but not regulated. A servo system was incorporated to keep the voltage to the motor constant. This produced adequate stability of the motor speed to levitate the sample for hours. For prudence, the operator would periodically adjust the operating frequency based upon behavior of the sample.

For safety and noise reduction, the siren levitation system was enclosed in a sound attenuating enclosure fitted

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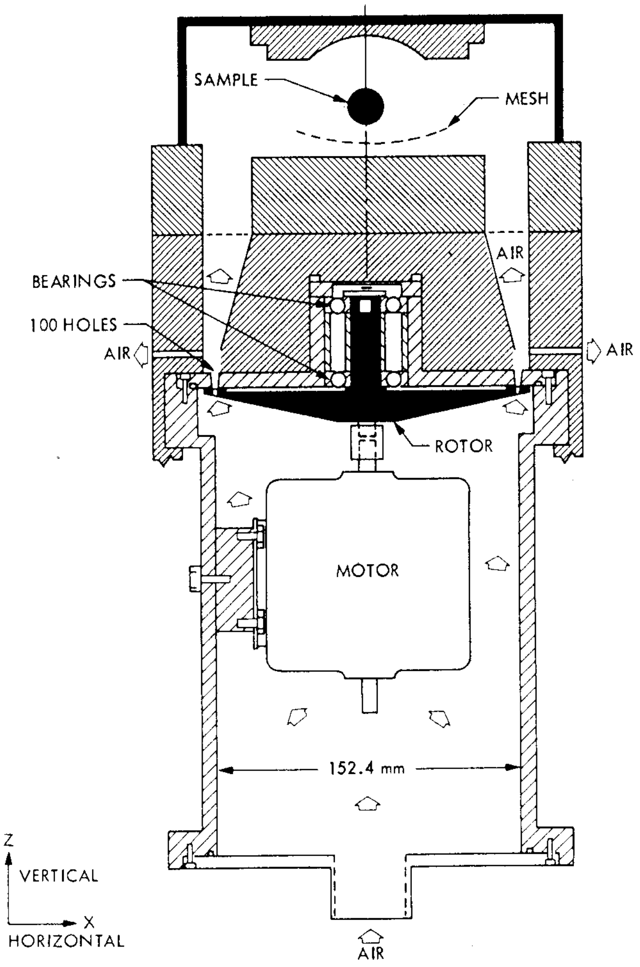

FIG. 1. Cross section of siren, motor, and reflector system showing direction of the air flow.

with a glass observation window. Even with the acoustic insulation, the sound levels were about 75 dB at 1 meter from the observation window.

The sample is positioned in the acoustic field by radiation pressure alone. This requires that standing waves be present which, in turn, necessitates some sort of reflector system. If the density of the sample is much greater than that of the medium, the radiation pressure is described by the Bernoulli equation:[2,8–10]

$$\langle P\rangle = (\overline{p^2}/2\rho c^2) - \tfrac{1}{2}\rho(\overline{V^2})$$

where $\langle P\rangle$ is the radiation pressure, p is the excess acoustic pressure, V is the gas particle velocity, ρ is the density of the gas, c is the velocity of sound in the gas, and a bar over a quantity denotes a time average.

A levitated sample will be driven towards a velocity antinode, where p^2 and the ambient pressure are at a minimum. On Earth, a balance is achieved between the weight of the sample and the acoustic force due to the radiation pressure. Since the pressure gradient and the force are zero at the reflector surface, the sample must initially be supported away from the surface, as by the mesh shown in Fig. 1.

II. MEASUREMENT TECHNIQUES

All sound-pressure measurements were made with 6.3-mm diameter Kistler 6001 quartz pressure transducers and a charge amplifier. The voltage from this amplifier was measured by a digital voltmeter and converted to decibels. The resolution of the sound-pressure level (SPL) measurements was ± 1 dB, relative to 2.0×10^{-4} dyne cm^{-2}. The noise level of the microphone and amplifier is 130 dB.

The choice of a microphone location that avoids perturbing the field appreciably is important. A transducer that was flush with the lower reflector was used for frequency measurements. For sound field mapping, the transducer was mounted on a 6.3-mm rod so the transducer and rod had the acoustic profile of a single rod.

III. EXPERIMENTAL OBSERVATIONS

Experiments included modifying the geometry of the reflector system, observing the effect of these changes on the position and stability of the sample, and measuring the sound field in order to better understand the factors affecting the quality of the levitation.

A. Frequency response and pressure field of the siren

The acoustical output of the siren, measured at the center of the flat reflector, with the upper reflector removed, was fairly constant over the frequency range of interest, varying only $\pm$ 2.2 dB from 2900 to 3600 Hz. The results are shown in Fig. 2. The acoustical pressure waveform displayed on an oscilloscope was seen to be amplitude modulated by 10% at around 140 Hz. Resonances in the lengths of tubing used to cool the air from the compressor are suspected.

The spatial distribution of the acoustical energy of the siren with only the matching horn is shown in Fig. 3. The siren was operated at 3260 Hz. The minima lie along a line that makes an angle of 0.68 rad (39°) with respect to the axis

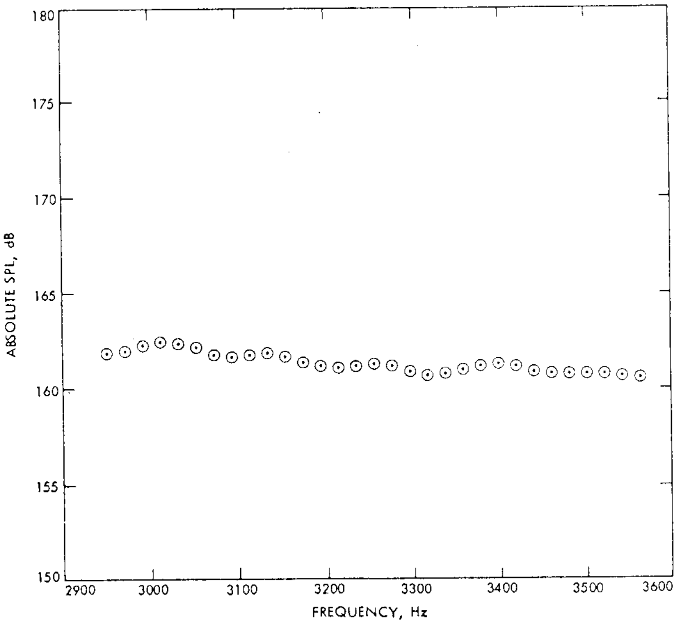

FIG. 2. Acoustic output versus frequency of the siren without a reflector. The microphone is located within the lower reflector, which is mounted atop the impedance matching horn.

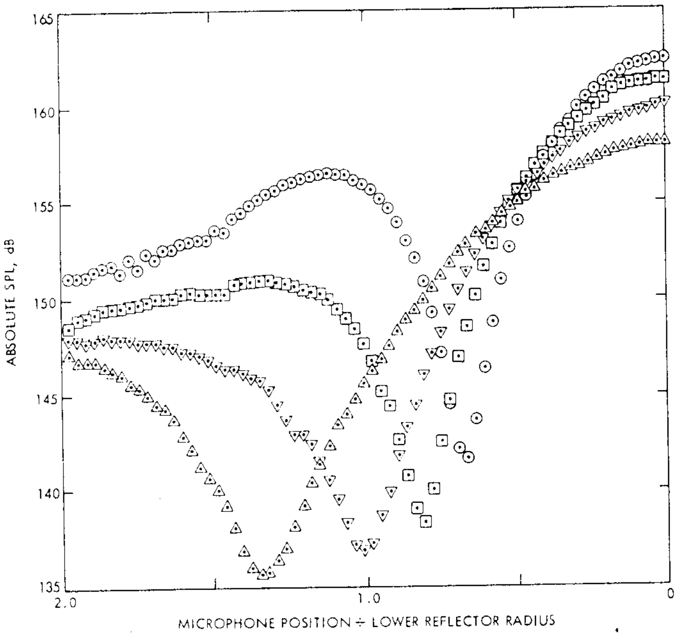

FIG. 3. Acoustic output versus horizontal position of the siren without reflectors at various heights (○ 25.4, □ 50.8, ▽ 76.2, and △ 101.6 mm) above the horn opening. The frequency is 3260 Hz.

of the siren and intersects near the outlet of the horn (67.6 mm above the siren holes). Allen and Rudnick observed the minima at approximately 0.56 rad (32°) at a radial distance of 25 cm for a similar siren operated at 3250 Hz with an exponential horn.[6] The horn used here had a straight taper. The present data are consistent with a ring source of 128 mm. The diameter of the lower reflector was 122 mm.

When the siren was used for levitation, a reflector was needed both to produce the standing waves and to further increase the power.

Two enclosed geometries, one cylindrical and one rectangular, were tried. With both geometries, the sound from the siren was concentrated by a horn and then was introduced through a small port in the center of the bottom wall. Both of the enclosed geometries resulted in very poor stability of the sample position.

Several open reflector geometries were tried: (1) two flat reflectors, (2) two concave reflectors, (3) a flat reflector on top and a concave reflector on the bottom, and (4) a concave reflector on the top and a flat reflector on the bottom. These geometries were investigated with various spacings between the reflectors and with several choices of the radii of curvature. The stability of the positioning was observed in the vicinity of the fundamental resonance frequency. Generally, the most stable levitation was obtained while operating slightly below that frequency. The three geometries that gave the most stable results are shown in Fig. 4. Of these three geometries, the modified concave reflector gave the best stability and the flat plate the least.

Experiments were conducted to determine the effects of the reflector spacing and geometry on the power and stability of the levitation. The lower reflector was 122 mm in diameter (0.725 of wavelength used). Several upper concave reflectors were tried, with different radii of curvature and distances between reflectors, and the siren frequency was varied for each configuration to optimize the strength and stability of the levitation. A spacing of 66 mm between 122-mm diameter flat reflectors was found to give stable levitation. A good choice of dimensions for the concave reflectors was a small (59-mm chord) portion of a hemisphere of 38.1-mm radius with a spacing of 56 mm between the outer edge of the upper concave reflector and the lower reflector. This 56-mm spacing was also used on the modified concave reflector system. These dimensions are in general agreement with the observation of Lee and Feng that the size of reflectors should be on the order of $\frac{2}{3}$ of a wavelength.[4]

Other geometries produced poorer results in that the sample moved more erratically in the horizontal plane. A full hemisphere of 38.1-mm radius gave excellent stability when levitating light samples, but could not lift a small steel ball. The small section of a hemisphere shown in Fig. 4(b) and (c) gave less stability for light samples but could levitate a single 19-mm diameter steel ball or three 12.7-mm diameter steel balls.

B. Observations of sample levitation

The system with a modified concave upper reflector [Fig. 4(c)] provided the most stable levitation of steel and lucite balls (of 12.7-mm and 19.05-mm diameters, respectively). The tendency of objects to spin typically enhanced the positional stability of symmetric samples. In the acoustic

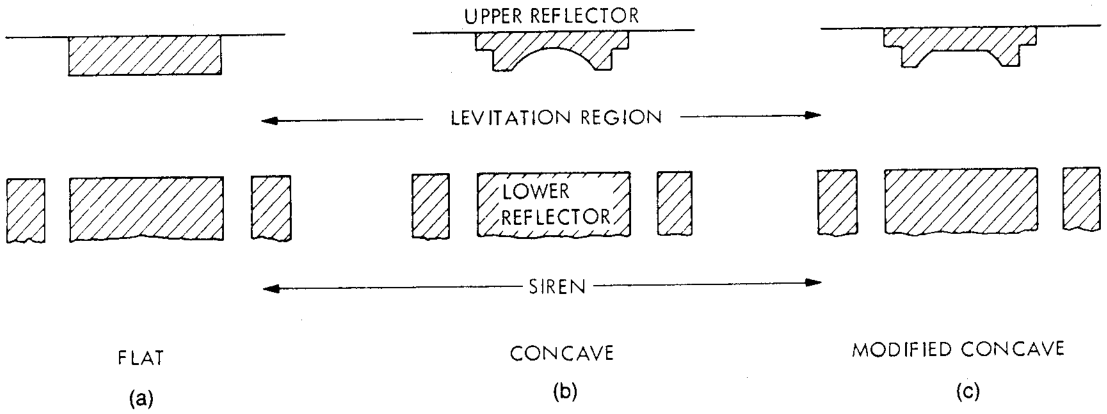

FIG. 4. Three of the more stable reflector geometries studied.

field of the siren–reflector system, the levitated object usually spun about a vertical axis. With a slight change of frequency or perturbation of the field (for example, introducing another object into the field), the spinning of the levitated object would often slow down, reverse direction, and speed up in the opposite direction. An irregularly shaped object, could be prevented from spinning by destroying the azimuthal symmetry of the field. Generally, the samples were levitated more stably when the siren frequency was slightly below the resonance of the reflector system.

Using the concave reflector geometry of Fig. 4(b) at a frequency below the resonance, the motion of the sample was usually very slight, being typically 4 mm horizontally and even less vertically. Increasing the frequency closer to resonance raised the sample but increased the motion. The most common motion observed in this and in other geometries was a downward facing arc. The maximum extent of the oscillatory motion was about 25 mm horizontally and 10 mm vertically. Operation at or above the resonance resulted in immediate instability. Other motions, such as vertical hopping, have also been observed.

C. Measurements of the sound field

Measurements made with each of the geometries of Fig. 4 included: the frequency spectrum at the center of the lower reflector and the spatial dependence of the SPL and waveforms in the levitation region. The effects of the presence of a sample and its position were also studied.

For each of the configurations of Fig. 4, there was a single resonance peak between 2950 and 3575 Hz. The Q (defined by the 3-dB points) was 29 when the flat reflector was used and 54 for the concave reflector geometries. The SPL at resonance was 4 dB higher for Fig. 4(b), which is consistent with its greater levitating ability. The modification of the concave reflector changed the magnitude of the resonance peak by less than 0.5 dB and increased resonance frequency by less than 1%.

The shift of the resonance frequency as a function of the vertical position of the sample can be predicted from volume exclusion and scattering effects. The dependence of the resonance on the sample position has recently been investigated for a rectangular geometry.[11,12] It was not clear how these results applied to the open system used here, so the resonance frequency shift was investigated experimentally.

For the modified concave reflector, putting the sample in the center of the region in which it would levitate lowered the resonance by 20 Hz but did not affect its magnitude nor the Q appreciably. Moving the sample 13 mm to the side increased the resonance frequency by about 5 Hz over that at the center.

The introduction of the restrained sample reduced the Q from 29 to 23 for the flat reflector geometry but did not affect the amplitude. It shifted the resonance down by 28 Hz. Moving the restrained sample 25.4 mm horizontally from the center raised the resonant frequency by 20 Hz, while raising the Q from 23 to 27.

For the concave upper reflector, the resonant frequency shifted by less than 5 Hz, and the amplitude changed less than 0.5 dB, while the Q did not change measurably. One reason the curved reflectors provide better stability is that their resonances are less sensitive to the sample position.

TABLE I. Harmonic content of acoustic field, measured at lower reflector.

Conditions	Relative strength (dB below fundamental)			
	2nd	3rd	4th	5th
No upper reflector				
at levitation frequency[a]	21	18	30	>37
Flat reflector				
no sample				
at resonant frequency[a]	24	24	29	30
(mike height: 15.2 mm)	16	28	32	35
(mike height: 30.4 mm)	6	20	24	25
at levitation frequency[a]	30	28	25	28
sample at center				
at levitation frequency[a]	32	23	26	30
sample 12.7 mm from center				
at levitation frequency[a]	32	24	27	30
Modified concave upper reflector				
no sample				
at resonant frequency[a]	4	16	18	26
at levitation frequency[a]	18	15	40	26

[a] Measured at lower reflector.

D. Harmonic content

Table I shows the harmonic content of the sound-pressure waves measured with and without reflectors. Data were obtained both for the empty system and with a restrained 12.7-mm diameter steel ball at several locations.

Without a reflector, an almost triangular waveform was observed near the levitation frequency. All of the harmonics were at least 18 dB below the fundamental, which contained more than 98% of the energy.

Incorporating a reflector increased the level of the fundamental. Lowering the frequency, from the resonance peak to the frequency used for levitation, decreased the strength of the second and third harmonics by 4 to 6 dB for a flat reflector and decreased the second harmonic measured at the lower reflector by 14 dB for the concave reflector.

The theory of King[13] predicts the force will increase as the square of the pressure and inversely with the wavelength; therefore, the force due to the second harmonic (157 dB) is half of that due to the fundamental and it is opposite in direction. Consequently, the presence of the second harmonic can be expected to seriously affect positional stability in this system. Operating away from the resonant frequency decreases the energy in the harmonics and permits stable levitation.

Greater stability is also expected if the sample is kept away from the midplane, where the second harmonic predominates. It has been repeatedly observed in other acoustic levitation systems that the most stable levitation occurs when the sample is levitated just above the plane of maximum force, which is an eighth of a wavelength above the lower reflector.

E. Mapping the sound field

The general variation of the sound field intensity and the pressure waveforms at different locations using the concave reflector can be seen in Fig. 5.

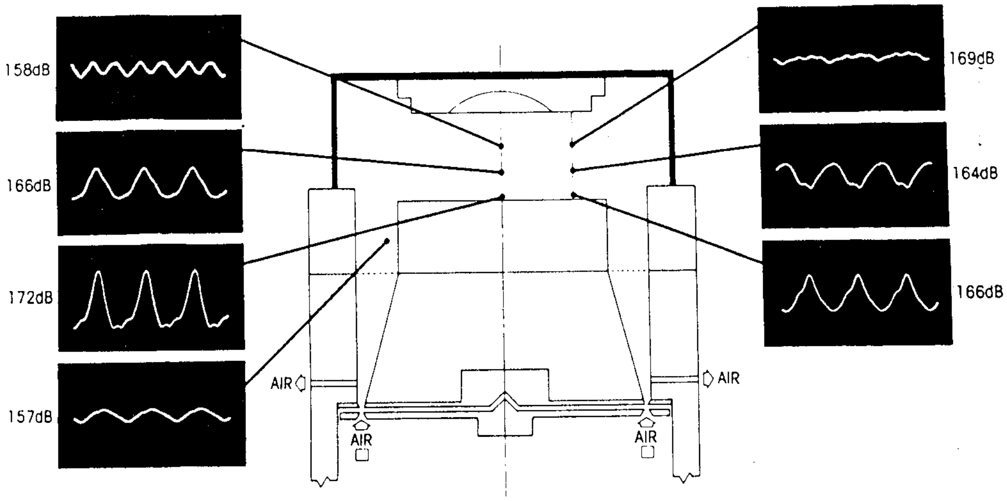

FIG. 5. Acoustic pressure waveforms in the siren–reflector system.

Figure 6 compares the distribution of the sound pressure for the flat reflector geometry to that for the modified concave reflector near the height of stable levitation. In order to rule out the possibility that the flatness of the curve for the concave reflector at the 12.7-mm elevation was due to the sound field saturating near 175 dB, the experiment was repeated, varying the pressure head of the air supply to the siren. The field plot retained its shape even when the siren output was reduced 13 dB by decreasing the pressure of the air supply from 5.2 to 2.1×10^4 N/m^2.

Data can be compared for these two geometries both near the plane of maximal force (i.e., 15.2 mm and 12.7 mm above the lower reflector for the flat and curved geometries, respectively) and near the plane of zero force (30.5 mm and 31.8 mm). A free sample levitates most stably 25 mm above the lower reflector. It can be seen that the curved reflector geometry gives a stronger and generally more uniform field in the region.

F. Effect of the sample and its position

The resonant frequency of a system is different when there is a sample and when there is none; in addition, the sample position can change the frequency by several percent. Leung *et al.* have demonstrated the relationship between the position of an object in a resonant system and the resonant frequency.[12] Even a frequency change of a few percent is critical when the Q of a system is high and the amplification it provides is required—as in a 1-g levitation system. Figure 7 shows the relation between sample position and the resonance frequency in a rectangular chamber; the data obtained by Leung corroborated this model. The analysis that they used for a closed system can be used to explain the enhanced stability obtained by operating the siren at a frequency below the empty-system resonance.

When levitating a sample in the laboratory, the acoustic and gravitational force balance, and the resulting sample position is slightly below halfway between the reflectors. As shown in Fig. 7, the change in the frequency of the resonance due to the sample location is negative. If the empty-system frequency was used with the sample there, it would be above the actual resonance. If the sample moved away from the center, the resonant frequency would increase, bringing it closer to the driving frequency, and the effective force would increase.

Unfortunately, this increased force lags behind the changing sample position because of the delay time imposed by the Q of the system. This increased force contributes little to slowing the sample as it moves away from its equilibrium position, but does push the sample back towards and

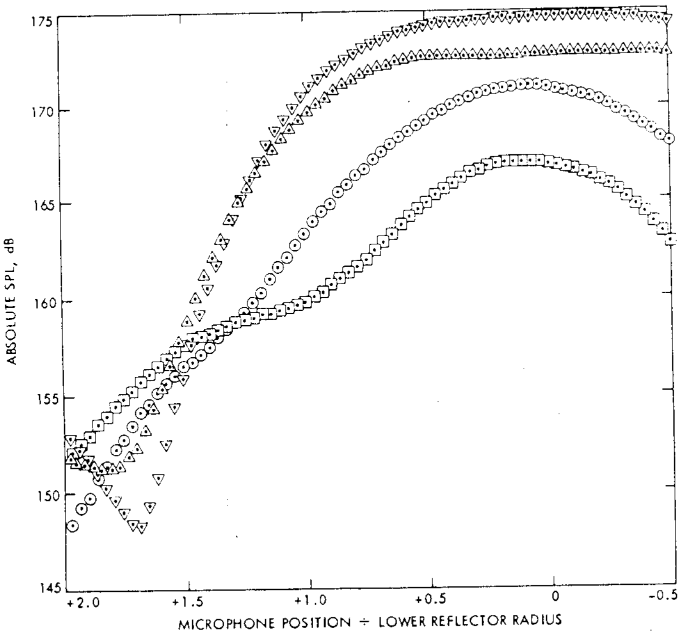

FIG. 6. Comparison of the acoustic fields produced by the flat reflectors and the modified concave upper reflector at various heights (○ 15.2, □ 22.9 mm, for the former, and ▽ 12.7, and △ 19.1 mm, for the latter) above the lower reflector.

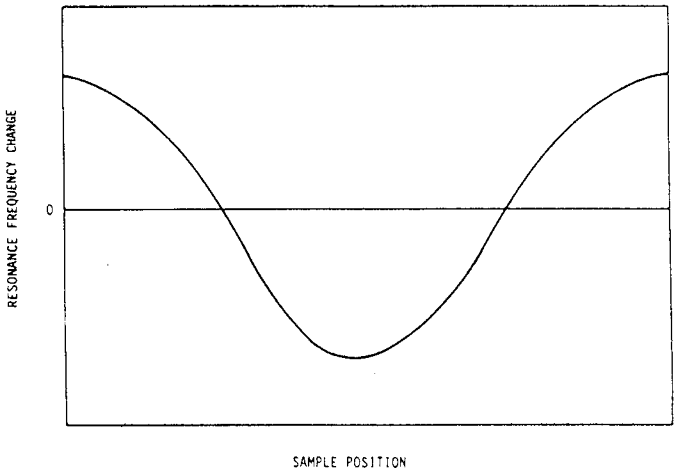

FIG. 7. The effect of the position of an object in a resonant system upon the frequency of the resonance. Data for a 2.5-cm sphere in a rectangular three-axis levitator ($15\times 13\times 13$ cm) were consistent with this model.[12]

through it with increasing force. The higher the Q of the system, the greater the lag and the greater the push back. The amplitude of the sample's excursions increases as energy is fed back into the oscillation. In contrast, for a system operated at a frequency that takes into account the presence of the sample, the force decreased when the sample moves away from the equilirium point. As it reaches its maximum excursion, the force decreases so the sample returns slowly, and the amplitude of the oscillations about the equilibrium point decreases.

A sample experiences a deeper well as it moves from its equilibrium position and a more shallow one as it returns to it. Analytical models are being tested of the dynamics connecting the sample position, the positioning frequency, and the resulting stability.[14]

G. Effects of heating the sample

Although the modified concave reflector gave excellent stability and could levitate heavy samples at room temperature, problems were encountered when localized heating was combined with levitation. Several attempts were made to heat the sample with a light source, by electrically heating the acoustic reflectors, and by supplying heat from a hot air jet that was an integral part of the lower reflector. In all cases, the acoustic field carried that heat away rapidly. Furthermore, the temperature gradients increased the stability problems. No reliable technique was found for heating a sample to high temperature ($>400\,^{\circ}$C) while levitating it with a siren.

IV. CONCLUSIONS

This study has demonstrated the feasibility of using a siren as the sound source for the stable acoustic levitation of high-density objects of centimeter size in the Earth's gravitational field.

The stability of levitated samples (centimeter-sized steel bearings and nonspherical glass shapes) was studied using various reflectors. The reflector was positioned above the siren. This amplified the acoustic radiation pressure to support the object's weight and to provide a small lateral centering force. Various geometries were used; the best results were obtained with a concave reflector. Its size and distance above the impedance-matching horn on top of the siren were roughly half of the wavelength at which the siren was operated.

The siren was operated at a frequency lower than the resonance of the chamber. Operating away from the empty-system resonance decreased the energy present in the higher harmonic—especially the second. Operating at a lowered frequency, the interaction between the sample's position and the acoustic restoring force decreased the amplitude of oscillations and enhanced stability.

Using the optimal reflector and adjusting the frequency, stable levitation was maintained for hours.

ACKNOWLEDGMENTS

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[13] L. V. King, "On the acoustic radiation pressure on spheres," Proc. R. Soc. London Ser. A **147**, 212–240 (1934).

[14] S. L. Garrett and M. Barmatz, "Detuning/dephasing instabilities in acoustic levitation and Rayleigh disk systems," J. Acoust. Soc. Am. Suppl. 1 **77**, S21 (1985).

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A Fifty Horsepower Siren

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This paper contains a description of the theory, construction, and testing of a siren with an acoustic output of fifty horsepower, at an efficiency of seventy percent. The siren is of the traditional type in which the air stream from a compressor passes through an orifice which is opened and closed at an audiofrequency. Part I contains a development of the theory on which the design was based. The theory is developed with the aid of an equivalent circuit diagram (Fig. 1) in which the various acoustic impedances are represented as lumped circuit elements. The theory is for the most part based on the customary linear equations for sound propagation, but the non-linear nature of the impedance of the port is taken into account. Mere inspection of the equivalent circuit diagram is sufficient for the understanding of the considerations which are important for the obtaining of a high efficiency. According to this theory the best efficiency can be realized in a siren when the orifice is provided with a suitable acoustic horn, when the excess pressure of the air supplied by the compressor is small compared with the absolute pressure of the atmosphere, and when the operations of opening and closing the port occupy a small fraction of the period of one cycle. Expressions are obtained for the theoretical efficiencies of several types of sirens, and are found to be near one hundred percent for a good design. Part II contains a description of the siren which was constructed to operate with a compressor that supplied 2500 cubic feet of air per minute at a pressure of five pounds per square inch. The siren contains six ports with a total area of 22 square inches, which are opened and closed by a rotary chopper at a frequency of about 500 cycles per second. Each of the ports is provided with an exponential horn whose cut-off frequency is 125 cycles per second. The theoretical efficiency of this siren is computed to be between 70 and 90 percent. Part III is devoted to a description of the testing of the siren and a discussion of the results obtained. The tests indicated that at full output the siren produces an average level of 138 db at a distance of 100 feet. The measured level at the mouth of the horns is 174 db, and the calculated level in the throat of the horns is 184 db above 10^{-16} watt per square centimeter. The level in the throat of the horns corresponds to a sound intensity of 260 watts per square centimeter. The most reliable method of determining the efficiency, based on thermodynamic principles, indicates that the efficiency of the utilization of the available energy in the air from the compressor is about 72 percent.

INTRODUCTION

THE purpose of this paper is to describe the construction and design principles of a highly efficient, fifty horsepower siren[1] which was developed by Bell Telephone Laboratories under the auspices of the National Defense Research Committee and in cooperation with the U. S. Army.

When the need for an efficient air-raid warning system became apparent toward the end of 1941, the Office of Civilian Defense obtained a release of the project from its classified category. Tests previously reported in this Journal[2] were made of the usefulness of the siren for this purpose in New York City. Further work on the siren was carried on by the Chrysler Corporation, who made important improvements in the device, particularly in respect to its serviceability and construction. These improved sirens were subsequently marketed commercially by that company. The description here, however, will be restricted to the device as developed for the initial New York City tests.

The project of designing the siren was under the general direction of E. C. Wente. A thoroughgoing theoretical examination of the problem which preceded the mechanical design was carried out by H. T. O'Neil and the author. The mechanical design was accomplished by T. L. Dowey, and the siren was assembled by Dowey and A. H. Muller. The field tests on the siren were made under the direction of F. K. Harvey and the author.

PART I. THEORETICAL CONSIDERATIONS

Because the prior art afforded no example of a siren with an efficiency greater than one or two percent, it was felt necessary to make a careful theoretical analysis of the problem of designing a siren which would have an efficiency of fifty percent or more.

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* Now with the Polaroid Corporation, Cambridge, Massachusetts.

[1] R. Clark Jones, J. Acous. Soc. Am. **14**, 129(A) (1942).

[2] J. B. Kelley, J. Acous. Soc. Am. **14**, 129(A) (1942).

It was known at the beginning of the theoretical investigation that the siren would be powered with a compressor which supplied 2500 cubic feet per minute at a pressure of five pounds per square inch. Although the theoretical treatments given below were made as general as seemed useful, the various approximations were governed by a consideration of the compressor to be used.

The type of siren to be considered here is the type in which the compressor delivers air to a pressure chamber which is connected to one or more horns or orifices by ports or openings whose area may be varied periodically.

Although an effort will be made to take into account the effects of non-linearity where they seem important, the analysis will be based essentially upon the usual linear equations for the propagation of sound waves.

The following list includes the more important symbols used in this report:

ω = angular frequency,
ρ = density,
c = speed of sound,
$k=\omega/c=2\pi$ divided by the wave-length,
γ = ratio of specific heats,
T = absolute temperature,
$V=i$ = volume velocity,
P = absolute pressure of the atmosphere,
$p=E$ = excess pressure maintained in the pressure chamber by the compressor,
$f=p/P$,
S = area of the horn throat,
A = instantaneous area of the port opening,
R, H, G = acoustic resistances, and
L, M = acoustic inductances.

Equivalent Circuit Diagram of the Siren

An approximate equivalent circuit diagram of a siren is shown in Fig. 1. The diagram is based on the convention in which the air pressure is represented by voltage, and the volume velocity of the air by an electrical current.

The combination of the resistance R and the inductance L in parallel represents the acoustic impedance of the throat of the horn or, if no horn is used, of the opening into the atmosphere. The combination of the resistance H and the inductance M in parallel represents the acoustic impedance of the opening into the pressure chamber. It will be supposed in the mathematical treatment that the pressure chamber possesses perfectly absorbing walls, so that the impedance which is seen by looking into the pressure chamber from the horn throat is simply that of an orifice in an infinite baffle. The variable resistance G represents the resistance of the device which controls the flow of air into the horn from the pressure chamber. Finally, the battery E represents the pressure maintained in the pressure chamber by the compressor.

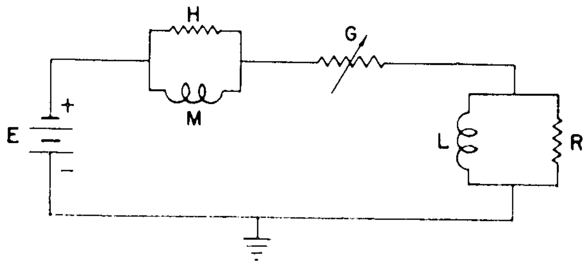

FIG. 1. The equivalent circuit which is used as the basis for the theory of the siren. The battery E represents the compressor; the resistance R and the inductance L in parallel represent the impedance looking into the throat of the horn; the elements H and M similarly represent the impedance looking back into the pressure chamber from the port; and the variable, non-linear resistance G represents the impedance of the port itself. The voltage with respect to ground corresponds to the pressure with respect to atmospheric pressure, and the current corresponds to the volume velocity of the air passing through the siren.

Qualitative Considerations

The considerations which result from th detailed analysis may be indicated in a qualitative manner by mere inspection of the equivalen circuit diagram.

If the siren is to have a high efficiency th radiation resistance of the horn, i.e., the serie resistance component of the parallel L and elements, should be large compared with th other series resistance components that are i the circuit when the ports are open, since th power which is radiated as sound is the powe which is developed in this resistance.

This fact has four important consequences:

(1) A horn is in general of great benefit and it be of the exponential type its cut-off frequenc should be small compared with the operatin frequency of the siren.

(2) It is always wasteful to control a flow, be it of fluid or electricity, with a variable resistance. There are, however, two values of series resistance which involve no waste of power zero and infinity. According the chopper whic

alternately opens nd closes the port at the throat of the horn ould be arranged so that the period of opening closing is small compared with the time the rts are open or closed.

(3) As will be s wn later in this report, the resistance of the rt when it is open is non-inear, its value ing approximately propor-ional to the veloc of flow. The excess chamber pressure p should erefore be small compared with the pressure of the atmosphere, if much of the energy is no be dissipated by turbulence n the port.

(4) The reacta of M should be small com-pared with the re ance H. This requirement is equivalent to th ondition that k times the maximum dimen n of the port be small com-pared with un here k is 2π divided by the wave-length.

Quantitative E ssions for the Impedances in Fig. 1

The impeda used throughout this report are acoustical edances, defined as the ratio of the pressure the *volume* velocity.

In Fig. 1 th oad impedance of the horn is represented by inductance and a resistance in parallel. In the athematical treatment of this circuit it will assumed that L and R have constant value for all frequencies, including direct current. tually, they are constant only when the port terminated by a conical horn. The solution c thus be applied strictly only to a horn of this pe and particularly not to an exponential ho which according to the familiar theory does no ansmit waves below the cut-off frequency and s an infinite impedance at zero frequency. Pr ded, however, that the fre-quency at whi the port is modulated is well above the cut- f frequency of the exponential horn, the elem ts L and R are essentially con-stant in this fr uency region. There is no need to be concern about the theoretical value of impedance of exponential horn at zero fre-quency, becaus t is known from experience that a steady curre of air will flow about as freely through an ex nential horn as it would if it were conical.

The theory ill be developed as though the horn were con al, and it will then be assumed that the theo applies also to an exponential horn with the same values of L and R at the fundamental frequency.

The chief advantage of the exponential over the conical horn is that, for a given length and for given mouth and throat areas, it has a higher value of L/R.

As one may easily verify by reference to any standard textbook on acoustics, the following relations hold for a conical horn

$$R=\frac{\rho c}{S};\quad L=\frac{\rho l}{S};\quad Q_1\equiv\frac{\omega L}{R}=kl, \tag{1}$$

and similarly for an exponential horn:

$$R=\frac{\rho c}{S}\left(1-\frac{\omega_0^2}{\omega^2}\right)^{-\frac{1}{2}};\quad L=\frac{\rho c}{\omega_0 S};$$

$$Q=\frac{\omega L}{R}=\left(\frac{\omega^2}{\omega_0^2}-1\right)^{\frac{1}{2}}, \tag{2}$$

where l is the distance from the throat to the virtual apex of the cone, S is the area of the horn throat, and ω_0 is the cut-off frequency of the exponential horn.

An estimate of the values of H and M may be obtained on the assumption that the impedance of the orifice is that of a circular piston of the same area. When the customary expressions for the impedance of a unilaterally-loaded piston in an infinite baffle are transformed in the manner required to determine the values of resistance and inductance in a parallel combination, involved expressions result, but if k times the radius, a, of the piston is small compared with unity, the expressions are represented approximately by

$$H=\frac{128}{9\pi^2}\cdot\frac{\rho c}{S},\quad M=\frac{8}{3\pi}\cdot\frac{\rho a}{S}, \tag{3}$$

so that

$$Q_2\equiv\frac{\omega M}{H}=\frac{3\pi}{16}ka. \tag{4}$$

It is more difficult to make a reliable estimate of the value of the resistance G, which at any instant depends upon both the flow of air through the port of the siren, and the port area. A rough estimate is obtained in the following manner.

When gas escapes through an orifice from a chamber in which the absolute pressure is P_0 into a chamber where the pressure is P_1, the rate of discharge of mass is[3]

$$c_0\rho_0 A\left(\frac{2}{\gamma-1}\right)^{\frac{1}{2}}\left[\left(\frac{P_1}{P_0}\right)^{2/\gamma}-\left(\frac{P_1}{P_0}\right)^{(\gamma+1)/\gamma}\right]^{\frac{1}{2}}, \qquad (5)$$

where A is here the area of the *vena contracta*. When the pressure difference p is small compared with the absolute pressure, (5) may be used to obtain

$$p = P_0 - P_1 = \tfrac{1}{2}\rho u^2,$$

where u is the *linear* velocity in the *vena contracta*. Applying this expression to the siren port, one obtains for its acoustical resistance

$$G = \frac{p}{V} = \frac{p}{Au} = \frac{1}{2}\rho\frac{|V|}{A^2}, \qquad (6)$$

where V is the instantaneous volume velocity through the port. In a study of the properties of Helmholts resonators L. J. Sivian[4] had to postulate a resistance term of the form (6) in order to account for the high damping which was observed when the resonators were oscillating at very large amplitudes.

A consideration of the mechanism of this resistance suggests that a necessary condition for its existence is the presence of a large and sudden increase in the cross-sectional area of the path through which the gas is flowing. Equation (6), therefore, gives a value of G that is too large when a horn is employed and the port is completely open, but a reasonably good value for the case in which the port is partially occluded. In the absence of a better estimate, the expression (6) will be used in the mathematical analysis, with A taken to be the instantaneous area of the opening.

Suitable expressions for the circuit elements of Fig. 1 having been obtained, it is now possible to derive expressions for the efficiency and sound output of any siren that can be represented by this circuit. Two cases will be considered, one in which the port area is varied with time in such a way that a pure tone is generated, and the other in which the operating times of opening and closing the port are a small fraction of the period of one cycle.

Theoretical Efficiencies

Siren with Pure Tone Output

Let it be required that the wave form of the acoustical output of the siren be sinusoidal. This is equivalent to the requirement that the current i_R through the resistance R be of the form

$$i_R = I\cos\omega t.$$

Considering the fact that L has zero impedance for direct current and that therefore no direct current flows through R, one finds the currents through the other elements of the circuit:

$$i_L = I_0 + (I/Q_1)\sin\omega t,$$
$$i_G = I_0 + I\cos\omega t + (I/Q_1)\sin\omega t,$$
$$i_H = Q_2 I\cos\omega t - \alpha Q_2 I\sin\omega t,$$

and

$$i_M = I_0 + \alpha I\cos\omega t + \beta I\sin\omega t,$$

where

$$\alpha \equiv \frac{-Q_2/Q_1}{1+Q_2^2}, \quad \beta \equiv \frac{Q_2 + 1/Q_1}{1+Q_2^2},$$

and where I_0 is the average value of the current through M, G, and L.

These relations serve to evaluate G, since, by definition, G is equal to the ratio of the voltage across G to the current through G:

$$G = \frac{E - Ri_R - Hi_H}{i_G}.$$

By Eq. (6), however, the resistance G may also be written

$$G = \frac{\rho i_G}{2A^2}.$$

Upon eliminating G between the last two relations, and solving for A, one finds

$$A = \left(\frac{\rho}{2}\right)^{\frac{1}{2}}\frac{I_0 + I\cos\omega t + (I/Q_1)\sin\omega t}{(E - IR(1+\beta Q_3)\cos\omega t + IR\alpha Q_3\sin\omega t)^{\frac{1}{2}}}, \qquad (7)$$

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[3] Horace Lamb, *Hydrodynamics* (Cambridge University Press, New York, 1932), sixth edition, p. 7.

[4] L. J. Sivian, J. Acous. Soc. Am. **7**, 94–101 (1935).

wh

$$Q_3 \equiv \omega M/R. \tag{8}$$

i rder that the minimum value of A be zero, it ecessary that the following relation hold bet en I and I_0:

$$I_0 = \kappa I, \tag{9}$$

wh

$$\kappa \equiv (1 + 1/Q_1^2)^{\frac{1}{2}}.$$

Th aximum value of A must be equal to the are S of the throat of the horn. Unfortunately, it i ifficult to determine exactly the maximum val of (7). In cases of practical interest, howeve in which Q_1 will be large, and in which Q_2 anc Q_3 will be small compared with unity, the ma num value of A will be very close to its val for $t=0$. Thus

$$S = \left(\frac{\rho}{2}\right)^{\frac{1}{2}} \frac{I(1+\kappa)}{(E - IR(1+3Q_3))^{\frac{1}{2}}}$$

app ximately. Upon solving this equation for I one ıds

$$I = (E/R\mu)F(y), \tag{10}$$

whe

$$\mu \equiv 1 + Q_3 \frac{Q_2 + 1/Q_1}{1 + Q_2^2},$$

and here

$$F(y) \equiv [(1+2y)^{\frac{1}{2}} - 1]/y,$$

wit

$$y \equiv \frac{(1+\kappa)^2}{\mu^2} \cdot \frac{\rho E}{S^2 R^2}.$$

Bot the dimensionless parameter y and the func on $F(y)$ are of fundamental importance in the eory of the siren as developed here, and are scussed in the following two paragraphs.

By use of the relation $\rho c^2 = \gamma P$ and Eq. (1) for K the expression for y may be written

$$y = \frac{(1+\kappa)^2}{\mu^2} \cdot \frac{f}{\gamma} \cong \frac{4f}{\gamma},$$

where $f \equiv p/P$ and where the approximate equality holds when both κ and μ are nearly unity. Since γ is very nearly 1.4 for air, the parameter y is roughly three times the ratio of the excess chamber pressure to the absolute pressure of the atmosphere.

The following very brief table indicates the behavior of the function $F(y)$ when its argument lies between zero and ten:

y	$F(y)$
0.0	1.000
0.1	0.954
0.2	0.916
0.5	0.828
1.0	0.732
2.0	0.618
5.0	0.463
10.0	0.358

The following two series expansions are also helpful in understanding the nature of the function:

$$F(y) = 1 - \tfrac{1}{2}y + \tfrac{1}{2}y^2 - \cdots, \quad y \leq \tfrac{1}{2};$$

$$F(y) = \left(\frac{2}{y}\right)^{\frac{1}{2}}\left[1 - \frac{1}{(2y)^{\frac{1}{2}}} - \frac{1}{4y} - \cdots\right], \quad y \geq \tfrac{1}{2}.$$

The efficiency of the siren may now be calculated. The output power is $\frac{1}{2}I^2R$, and the input power is EI_0. The efficiency is therefore

$$\mathrm{Eff} = \frac{I^2R}{2EI_0} = \frac{IR}{2\kappa E} = \frac{1}{2\kappa\mu}F(y). \tag{11}$$

Because the parameters κ and μ are always greater than unity, and the function $F(y)$ approaches unity from beneath as y approaches zero, *the efficiency of the siren considered here will always be less than 50 percent.*

Siren of Maximum Power Output

The siren just discussed, whose port area varied in such a way that the acoustical output contained only the fundamental frequency, is not the most efficient type of siren. It has already been pointed out that it is inefficient to control a flow of air by means of a variable resistance. Accordingly, a type of siren will now be examined in which the port is fully open half of the time, and fully closed during the remaining half of the cycle.

Because of the mathematical complexities of handling non-linear circuits, it is not possible to provide for this case a simple treatment which takes into account all of the details which were included in the first calculation. The problem will be simplified to the point where a simple treatment is possible. It is necessary to assume that either H or M is equal to zero, and to assume that the inductance L is infinite. These assumptions have as their consequence

$$Q_1 = \infty, \quad \alpha Q_3 = \beta Q_3 = 0, \quad \kappa = \mu = 1.$$

With these simplifications the volume velocity through the port is constant during the time that the port is open, so that the impedance G has only two values, infinity and G_0, where G_0 remains to be determined.

The calculation is very simple. Let I_0 be the constant current through the inductance L. Then when the port is closed, the current through R must be $-I_0$. In order that the average current through G also be equal to I_0, it is necessary that the current through G be equal to $2I_0$ while the port is open, and therefore the current through R is I_0 while the port is open. The total potential difference across R and G is then

$$E = 2G_0I_0 + RI_0. \tag{12}$$

Because the volume velocity through the port, $2I_0$, is equal to U, Eq. (6) may be written

$$G_0 = \rho I_0/S^2.$$

Substitution of the last equation in (12) yields

$$I_0 = (E/R)F(y), \tag{13}$$

where y differs only slightly from the parameter indicated by the same symbol in the first calculation:

$$y = \frac{4\rho E}{R^2S^2} = \frac{4f}{\gamma}.$$

The efficiency may now be determined directly. The input power is EI_0, and the output power is I_0^2R; the efficiency is therefore RI_0/E:

$$\text{Eff} = F(y). \tag{14}$$

This is the efficiency when the *total* acoustic output is taken into consideration. Since the output signal has the form of a square wave, an appreciable fraction of the total power output will be in the form of harmonics of the fundamental frequency. Actually, a fraction $8/\pi^2$ of the power in a square wave lies in the fundamental component. Thus if one is interested only in the fundamental component in the output of the siren, the corresponding efficiency is

$$\text{Eff} = (8/\pi^2)F(y). \tag{15}$$

Comparison and Discussion

The siren considered in the first calculation has an efficiency whose upper limit is 50 percent. The siren of the second calculation, however, has an efficiency whose upper limit is either 100 percent, or 81.06 percent, depending on whether the total acoustic output, or merely the fundamental component, is used in computing the efficiency.

Perhaps the most striking aspect of the comparison of the two calculations is the fact that the efficiencies depend on the parameter y in the same manner. Indeed, if the same approximations are introduced into the result of the first calculation that was used through[illegible] the second, namely, Q_2 and Q_3 zero, and Q_1 infinite, then the expressions (11) and (14) for the efficiencies in the two cases become identical, with the important qualification that the result for the first case contains an extra factor of one-half.

This comparison suggests that a slight improvement in the accuracy of the result for the second case may be obtained by using for the second case the expression (11) with the factor one-half omitted, and with all of the Q's given their value at the fundamental frequency:

$$\text{Eff} = \frac{1}{\kappa\mu}F(y). \tag{16}$$

Power Considerations

It might at first be supposed that the input power for a given port area S would be less for the first type of siren than for the second, but this is not the case, for upon comparing Eqs. (9) and (10) with (13) when κ and μ are set equal to unity it is evident that the input currents are the same. Thus the difference between the two types lies in the power output, and not in the power input.

Thus in order to obtain from the treatment of the pure tone siren the input and output powers which correspond to the square-wave siren on the basis of the assumption involved in Eq. (16), the output power obtained in the treatment of the pure tone siren should be multiplied by two, whereas the input power should be left unchanged. This procedure is followed in the following statements of the output and input powers of the square-wave siren. Thus, in order to obtain the corresponding results for the pure tone siren, the output power given below should be divided by two.

Total Power Output

On the basis of the procedure just described, the total power output is twice $\frac{1}{2}I^2R$, or simply I^2R. If the value (10) is used for I, the total power output W_0 is given by:

$$W_0 = \frac{E^2}{\mu^2 R} F(y)$$

which by virtue of the relation $\rho c^2 = \gamma P$ and Eq. (1) for R, becomes

$$W_0 = \frac{p^2 Sc}{\gamma \mu^2 P} F(y).$$

The result when p is small compared with P is obtained directly by setting F equal to unity, whereas when p is large compared with P, the function $F(y)$ is approximately equal to $(2/y)^{\frac{1}{2}}$, whence

$$W_0 \cong \frac{2pSc}{(1+\kappa)^2}.$$

Thus in the latter case, the output power is directly proportional to p; whereas in the former case, it is proportional to the square of p.

Total Power Input

Similar expressions may be written for the total power input. The total power input W_i is equal to EI_0:

$$W_i = \frac{\kappa E^2}{\mu R} F(y) = \frac{\kappa p^2 Sc}{\gamma \mu P} F(y),$$

and the corresponding expression when p is large compared with p is

$$W_i \cong \frac{\kappa Sc}{1+\kappa} \left(\frac{2p^3}{\gamma p} \right)^{\frac{1}{2}}.$$

Thus W_i varies as the second or three-halves power of p when f is small or large compared with unity, respectively.

The Power Available from the Exhaust of a Compressor

In the preceding treatment, it was assumed throughout that the total power available from the compressor was $EI_0 = pV_0$. It was not made clear, however, at what point the volume velocity V_0 was to be measured, whether at the intake of the compressor, at its exhaust, or in the horn of the siren. As already pointed out, the mathematical treatment was based primarily on the customary linear equations for the propagation of sound, so that no account is taken of the variation of the average volume velocity in the various parts of the system.

It seems reasonable to choose the volume velocity V_0 so that the expression pV_0 is exactly equal to the power available from the compressor.

Let P be the absolute pressure at the intake of the compressor,* let $P_c = P + p$ be the absolute pressure at the exhaust of the compressor, and let V_a be the volume velocity at the intake of the compressor. Then if the air is compressed adiabatically, the power required by a compressor whose cycle of operation is reversible, is[5]

$$W = \frac{\gamma}{\gamma - 1} P V_a \left[\left(\frac{P_c}{P} \right)^{(\gamma-1)/\gamma} - 1 \right],$$

$$= \frac{\gamma}{f(\gamma-1)} p V_a [(1+f)^{(\gamma-1)/\gamma} - 1],$$

where $f = p/P$. The same amount of power may be obtained from the air at the exhaust of the compressor, provided that the ambient pressure is the same as that at the intake to the compressor, and provided that no heat is added to or removed from the air stream.

Thus in order that the expression pV_a be an exact expression for the available energy, the volume velocity V_0 must be given by

---

* In order to agree with the subscript convention (a for atmosphere, c for chamber) used in this section, the absolute pressure of the atmosphere should be denoted by P_a instead of P, but the latter symbol is used in order to agree with the remainder of the report.

[5] W. N. Barnard, F. O. Ellenwood, and C. E. Hirschfield, *Elements of Heat-Power Engineering* (John Wiley and Sons, New York, 1926), third edition, pp. 130, 131.

$$V_0=\frac{\gamma}{f(\gamma-1)}V_a[(1+f)^{(\gamma-1)/\gamma}-1],$$
$$=V_a\left(1-\frac{1}{2\gamma}f+\frac{\gamma+1}{6\gamma^2}f^2+\cdots\right). \tag{17}$$

If one assumes an ideal, adiabatic compressor, the relation between the volume velocity V_a at the intake and the volume velocity V_c at the exhaust is

$$V_c=(1+f)^{-1/\gamma}V_a$$

In terms of V_c, the energy available from the air stream is given by

$$V_0=\frac{\gamma}{f(\gamma-1)}V_c[1+f-(1+f)^{1/\gamma}],$$
$$=V_c\left(1+\frac{1}{2\gamma}f-\frac{2\gamma-1}{6\gamma^2}f^2+\cdots\right). \tag{18}$$

A mixed formula is also available

$$V_0=\frac{\gamma}{f(\gamma-1)}(V_aV_c)^{\frac{1}{2}}[(1+f)^{1-(1/2\gamma)}-(1+f)^{1/2\gamma}],$$
$$=(V_aV_c)^{\frac{1}{2}}\left(1-\frac{2\gamma-1}{24\gamma^2}f^2+\cdots\right).$$

Of these three equations for V_0 Eq. (18) is the only one which depends solely on the actual pressure and volume velocity of the air stream, and, accordingly, this formula is valid for all air streams, whether they be obtained from an ideal compressor or not. For $f=\frac{1}{3}$, and $\gamma=1.40$, the above expressions become

$$V_0=0.8995\,V_a,$$
$$=1.1047\,V_c,$$
$$=0.9968(V_aV_c)^{\frac{1}{2}}.$$

These results indicate that the mean proportional of V_a and V_c is very nearly equal to V_0 for values of f as small as those considered here, whereas the volume velocities V_a and V_c differ appreciably from V_0.

A Thermodynamic Method of Measuring the Efficiency

In addition to the direct methods of measuring the efficiency of a siren, which involve measuring both the input and the output powers, there exists also a thermodynamical method which depends on the following considerations: If all of the available energy were removed from the air in the pressure chamber, the temperature of the exhausted air would be less than the temperature of the gas in the chamber by the amount corresponding to an adiabatic, reversible expansion from the pressure P_c to the pressure P. On the other hand, if the air were to escape from the chamber in a free expansion so that no work were done, the temperature of the air would not decrease at all, except by a small amount because of the Joule-Thomson effect which may be completely ignored in the present connection; in this

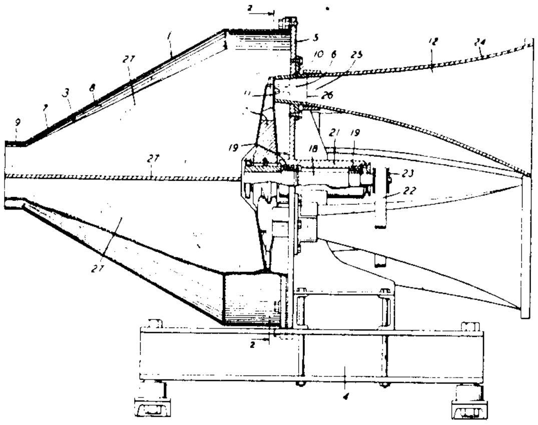

FIG. 2. A sketch showing a cross-sectional view of the siren. The rotor is seen edge-on, and is indicated by the number 2.

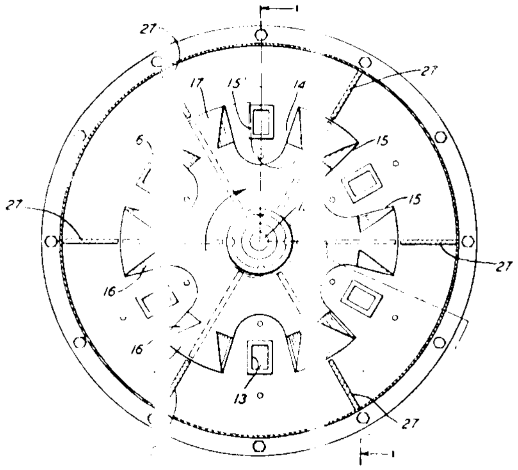

FIG. 3. A sketch showing the shape of the rotor and its position with respect to the six ports, one of which is indicated by the number 13.

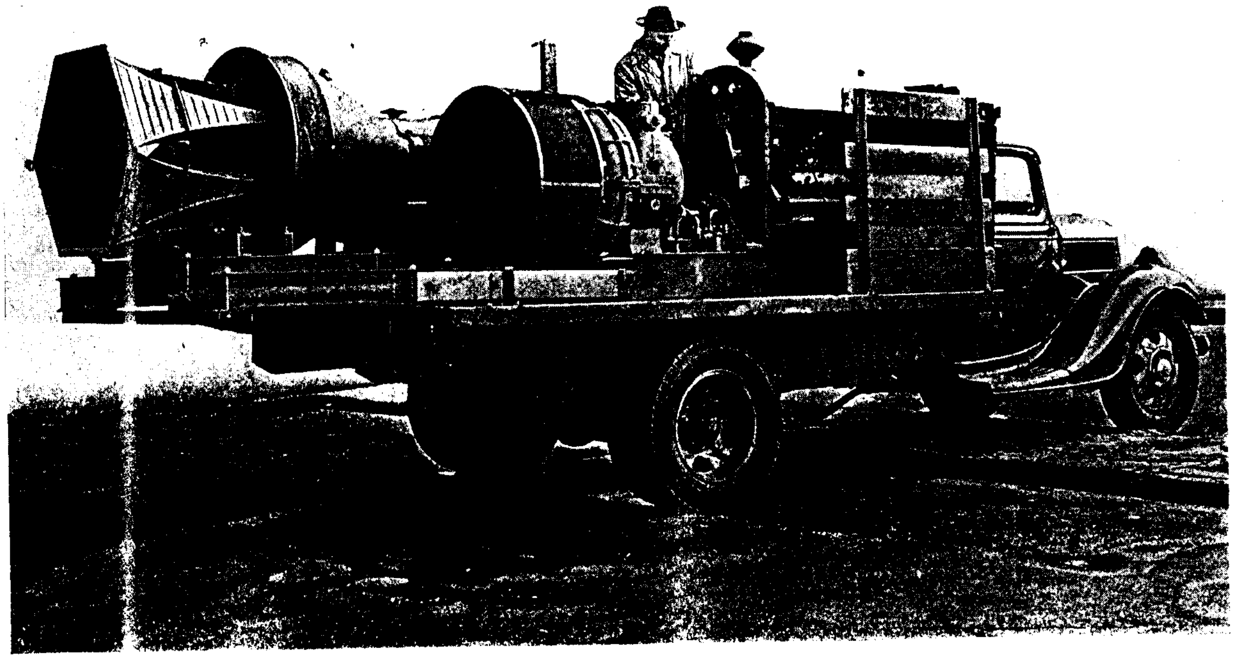

FIG. 4. View of the siren mounted on the truck. The intake filter and the compressor are shown to the right. The cone-shaped pressure chamber and the exponential horns on the left.

case all the available energy would be used to heat the gas.

Accordingly, the efficiency of the siren may be determined by measuring the temperature difference between the air in the horn, and the air in the pressure chamber. The efficiency is equal to the ratio of this temperature difference, to the temperature difference of an adiabatic reversible expansion from the pressure P_c in the chamber to the pressure P in the horn.

The difference between the absolute temperature T_c of the air in the pressure chamber and the temperature T_a of the air in the horns will be

$$
\begin{aligned}
T_c - T_a &= T_c\left[1-(1+f)^{-(\gamma-1)/\gamma}\right],\\
&= T_c\frac{f(\gamma-1)}{\gamma(f+1)}\left[1+\frac{1}{2\gamma}f-\frac{2\gamma-1}{6\gamma^2}f^2+\cdots\right],\\
&= T_c\frac{f(\gamma-1)}{\gamma(f+1)^{(1/2\gamma)}}\left[1-\frac{2\gamma-1}{24\gamma^2}f^2+\cdots\right],
\end{aligned}
\qquad (19)
$$

if all of the available energy is converted into a form of energy other than heat. If the assumption is made that the only other form of energy into which the available energy can be converted is acoustical energy, then the efficiency of the siren is the ratio of the measured difference between T_c and T_a to the value predicted by Eq. (19). For $f=\frac{1}{3}$, $\gamma=1.40$, the temperature difference calculated by (19) is

$$T_c - T_a = 0.0789T_c = 25.6^\circ\mathrm{C}, \quad T_c = 325^\circ\mathrm{K}.$$

This temperature difference is sufficiently large to be measured easily except in the case of very low efficiencies.

PART II. CONSTRUCTION OF THE SIREN

This section contains a description of the siren which was constructed to operate in conjunction with a compressor having a rated capacity of 2[illegible]00 cubic feet of air per minute at a pressure of 5 lb/in.^{2}. The frequency of operation was to be about 500 cycle/sec. The siren was designed on the basis of the theory given in the previous section.

It should perhaps be mentioned here that the siren was constructed at a time when there was some doubt that an efficiency as high as 50 percent could be realized; it was thought that possibly something important was being overlooked in the analysis given in the last section, because data available on previously constructed sirens indicated an efficiency not greater than one or two percent. Accordingly, the designers leaned over backwards to follow out all of the precepts of the theory.

Mechanical Design

The compressor used to supply compressed air for the siren was of the single-stage, centrifugal type. It was driven through step-up gears by a 95 horsepower gasoline engine. When operating at a speed of 10,000 r.p.m., it is capable of supplying air at an excess pressure of 5 lb./in.² when the volume velocity at the *intake* of the compressor is 2500 cubic feet per minute, or 41.67 cubic feet per second. This is the volume velocity denoted by V_a in the preceding theory. By Eq. (17), the effective value V_0 is equal to 37.45 cubic feet per second. At a pressure of 5 lb./in.², this flow represents 49.0 horsepower, or 36.6 kilowatts.

The general construction of the siren is shown in Figs. 2–4.

The air from the compressor enters the cone-shaped pressure chamber at the apex of the cone. A rotor equipped with six blades alternately closes and opens the ports at the throats of each of the six horns. All of the ports open at the same time and close at the same time.

The cylindrical section of the pressure chamber is 36 inches in diameter, but the inner diameter is only 30 inches because of a three-inch layer of rock wool which covers the inside of the pressure chamber. The rock wool was added in order to reduce standing waves within the chamber. It was not felt that the standing waves would seriously reduce the efficiency of the siren, but it was thought possible that they would make the performance of the siren change appreciably with small changes in the frequency of operation, so that it might be difficult to make reproducible measurements without careful control of the frequency.

The pressure chamber is provided with baffles at the places indicated by 27 in Figs. 2 and 3. These baffles serve to divide the pressure chamber into six parts, one for each port; their primary purpose is the prevention of a rotary motion of the air as a consequence of the rotation of the rotor.

The ports are rectangular, with dimensions $1\frac{3}{8}$ by $2\frac{2}{3}$ inches; the long dimension lies in the radial direction. The total area of the six ports is thus 22 in

The dimensions of the ports represent a compromise between a desire to keep the width of the port small in order to minimize the diameter of the rotor, and a desire to keep the length of the port small in order to minimize the radiation of sound back into the chamber.

The rotor is of cast duralumin, and is 25 inches in diameter. A clearance of 5 to 10 thousandths of an inch is maintained between the rotor and the ports. The centers of the ports are located $10\frac{2}{3}$ inches from the axis of the rotor; the circumference at this radius is thus 67.0 inches. The width of the blades and the width of the space between them are made equal, so that the width of the blades at the radius of $10\frac{2}{3}$ inches is one-twelfth of 67.0 inches, or 5.58 inches. This provides a ratio of 4.06 between the width of the blades and the width of the ports. Thus the ports are fully open during 38 percent and fully closed during an equal fraction of the period. The operations of opening and closing the ports each require 12 percent of the period.

The rotor is driven through five V-belts in parallel by a Wisconsin gasoline engine rated at 20 horsepower. It requires only 10 horsepower to turn the rotor at 4400 r.p.m. when the excess pressure in the chamber is zero, but the power required rises to 20 horsepower when the chamber pressure is 5 lb./in.²

The horns have an exponential taper, such that the cut-off frequency is 125 cycles per second. The rectangular shape of the horn at its throat is gradually transformed into an equilateral triangle as the horn flares. The six equilateral triangles are combined to form an equilateral hexagon, with 28 inches between opposite sides of the hexagon. Since the wave-length is 26.4 inches at 500 cycles per second, this diameter is sufficient to provide a resistive termination for the horn. In some of the tests of the siren, however, the directivity was increased by attaching a seven-foot wooden extension. This extension horn served to increase the size of the hexagon from 28 to 56 inches.

The siren was originally equipped with wooden exponential horns, but when these went to pieces after a few minutes of operation they were replaced with welded steel horns. The steel horns were fabricated from a double layer of 0.0508″ sheet steel. The two layers were tacked together with about one spot-weld per square inch. The purpose of the double layer is to deaden the sheet so that vibrations of large amplitude are not apt to occur.

Computed Performance

In the following paragraphs, the performance of the siren just described will be computed on the basis of the theory developed in Part I.

In order to calculate the theoretical performance of the siren, it is necessary first to calculate the dimensionless parameters κ, μ, and ν. These, in turn, depend in part on the values of Q_1, Q_2, and Q_3. The Q's will be computed for a frequency of 500 cycles per second.

The Q of the horn is very nearly equal to the ratio of the operating frequency to the cut-off frequency, according to (2). Since the cut-off frequency is 125 cycles per second, one has

$$Q_1 = 4.$$

If a is taken as the radius of a circle whose area is equal to 22.6 in.^{2}, one finds on substituting in (4)

$$Q_2 = 0.151$$

and from (2), (3), and (8)

$$Q_3 = \frac{8}{3\pi} ka \left(1 - \frac{\omega_0^2}{\omega^2}\right)^{\frac{1}{2}} = 0.21.$$

A more accurate calculation which takes into account the rectangular shape of the ports raises slightly the values of Q_2 and Q_3, but not enough to be important for the present computation.

With these values of the Q's, and with $p/P = \frac{1}{3}$, $\gamma = 1.40$, the parameters κ, μ, and ν may now be calculated:

$$\kappa = 1.031, \quad \mu = 1.085, \quad \nu = 0.834 \quad F = 0.759.$$

With these numerical values, the efficiency (16) is 0.678 and the volume velocity V_0, according to (10) is 28 ft.^{3}/sec. The last value is to be compared with the volume velocity V_0 which the compressor will supply: 37.45 ft.^{3}/sec.

Thus it would appear that the siren described here would not be able to utilize all of the capacity of the compressor, but considerable doubt exists as to the accuracy of the method by which the factor F was computed. This factor takes account of the existence of the non-linear resistance G; if G were zero, F would be equal to unity. It has already been pointed out that a necessary condition for the existence of the resistance G as defined by (6), is the presence of a large, sudden increase in the cross section through which the gas is flowing. In the siren described here, no such sudden increase exists during 75.2 percent of the time that the port is more than half open. Accordingly, it was decided that the best estimate of F would be much nearer to unity than to 0.76.

In the design the factor F was actually taken to be unity, so that the siren considered here was assumed to consume 38 ft.^{3}/sec. at a pressure of 4.9 lb./in.^{2} This decision was made in the light of the fact that there would be a small leakage of air through the ten thousandths of an inch clearance between the rotor and the ports. A rough estimate of this leakage suggested that it might be as high as three or four percent of the total flow available from the compressor. It should be noted that the pressure difference between the two sides of the blade, during the time that the port is closed, is more nearly equal to ten pounts per square inch than to five.

The above argument thus suggests that the siren here described will consume 38 ft.^{3}/sec. at a chamber pressure of 4.9 lb/in.^{2}, and that the efficiency of the utilization of the energy available in the air flow will lie between 70 and 90 percent.

PART III. TESTING OF THE SIREN

The quantitative tests of the siren were made on the western shore of Barnegat Bay where it could be pointed over a clear stretch of six miles of water. During all of the tests, the siren was mounted on the truck as shown in Fig. 4.

It was at the initial test of the siren in December, 1941, that the original wooden horns proved

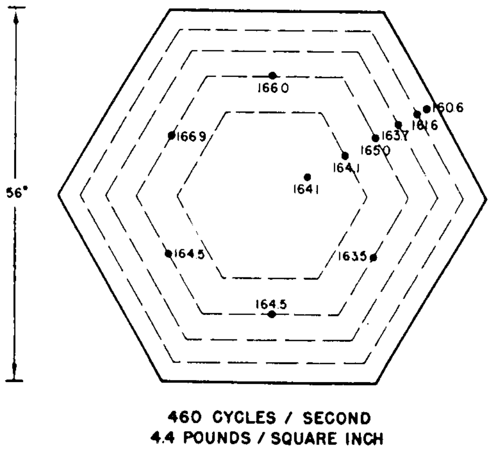

FIG. 5. A plot showing the results of one set of measurements of the sound level at a number of points in the plane of the mouth of the extension wooden horn. The areas between each pair of adjacent hexagons are equal. The points at which the measurements were made were chosen so that the results could be integrated to determine the total output power.

so completely inadequate. These horns took the place of the steel exponential horns which are visible in the photographs. They were constructed of half-inch plywood, with all the joints glued and screwed, and were reinforced with wooden cleats which were bolted together.

The siren was operated at full output for a period of about five minutes at a frequency of 440 cycles per second. When the test was over it was found that the ground beneath the wooden horns was littered with more than a hundred screws, nuts, bolts, and fragments of wood. Nearly all of the glued joints had opened up, and in doing so, had either turned out or pulled out the wooden screws. Most of the cleats had come loose, and in some places fragments of the wood had splintered off.

As the result of this experience, new horns were constructed of sheet steel as previously mentioned.

The principal objectives in these tests were the determination of the efficiency and the sound output.

Efficiency and Acoustic Output

Efficiency by the Thermodynamic Method

This method was described in the first section of this report, and is the only method which determines the efficiency without separate measurements of the input and output powers. It involves merely the measurement of temperatures.

Unfortunately, the application of this method to the siren described here presented a difficulty. The power required to drive the rotor (20 horsepower when the chamber pressure is 5 lb./in.^{2}, but only 10 horsepower when the chamber pressure is zero) is not insignificant compared with the total of 49 horsepower in the air stream. This power, by heating the air stream, reduces the temperature difference on which the determination of the efficiency is based. To be sure, the correction is readily calculated if one assumes that all of the energy is converted into heat. This assumption is somewhat questionable since the power required to drive the rotor doubles when the chamber pressure rises to its operating value.

The calculation of the efficiency by the thermodynamic method is based on the following measurements:

Chamber pressure: 4.7 lb./in.^{2},
Frequency: 465 cycles per second,
Ambient temperature: 2°C,
Temperature difference between chamber and air in horns: 15°C.

On the assumption that the absolute temperature of the air in the chamber pressure was 310°K, the temperature difference calculated by Eq. (19) is 24.5°C. Thus the apparent efficiency is only 61 percent.

No account, however, has yet been taken of the effect of the rotor. If all of the 20 horsepower used to drive the rotor were converted into heat, it would increase the temperature of the air in the horn 9.5°C, and the calculated efficiency would be just one hundred percent. If only half of the energy put into the rotor were converted into heat, the efficiency would be 81 percent with respect to utilization of the available energy from the compressor, and would be 50 percent with respect to the utilization of the energy put into the rotor. If, finally, the efficiencies of utilization of the two sources of energy were equal, the efficiency would be 72 percent, which is the result which is obtained from (20) if both the compressor and the rotor were considered as sources of input power.

Mapping Method

The second method used to determine the efficiency was to measure the acoustic level of the fundamental component at a number of points distributed over the plane of the mouth of the horn. The measurements were made at the mouth of the seven-foot extension horn, where the sound field was more uniform than at the mouths of the exponential horns.

The sound level measurements were made with a Western Electric 640-A condenser microphone. To avoid serious overloading of this microphone, the diaphragm was shielded with a steel cover by which the sound level was attenuated 36.0 db at frequencies below 1000 cycles per second. The resonant frequency of the steel cover was about 2000 cycles per second. The microphone with its shield was calibrated by a free-field comparison with a calibrated 640-A microphone.

The pressure levels shown in Fig. 5 were obtained with a chamber pressure of 4.4 lb./in.^{2} at an operating frequency of 460 cycles. The zero reference level is the customary one of 10^{-16} watt/cm^2 in a plane progressing wave.

The levels were converted to non-logarithmic power units and were integrated over the surface of the opening of the horn. The result obtained corresponded to an average level of 164.1 db over the entire surface of the opening, or 2.57 watt/cm^2. The total area of this surface is 17,520 cm^2. The total power is therefore 45.0 kilowatts, or 60.3 horsepower.

If the total input power is taken to be 49 plus 20 horsepower, the efficiency is 87.4 percent.

The output power just computed, however, refers only to the fundamental component. Although no mapping was attempted for the harmonics, the average of measurements at several points indicated that the level of the second harmonic was seven decibels less than that of the fundamental, and the level of the third harmonic, fourteen decibels less than that of the fundamental. The inclusion of these two harmonics serves to increase the output power by the factor 1.24. (If the output were a square wave, the second and third harmonics would be down 9.5 and 14.0, respectively, and the inclusion of *all* of the harmonics would increase the output power by the factor $\pi^2/8 = 1.234$.)

The inclusion of the second and third harmonics thus serves to increase the measured output power to 75 horsepower, which corresponds to an efficiency of 108.8 percent!

This result is certainly too high. A reduction of only 1.7 db in the average level, however, would suffice to reduce the efficiency to 75 percent. It is not difficult to believe that an error of this magnitude could be present in the measurement. In addition to questions of temperature stability of the shielded microphone, there is firstly, the fact that there is evident in Fig. 8 a difference of as much as 3.4 db in the levels at points where the levels should be equal on the basis of symmetry considerations; a considerable error could therefore enter in the process of integration over the mouth of the horn. Secondly, the measured level fluctuated irregularly over a range of about two decibels.

Accordingly, it may be said in conclusion that the efficiency obtained by a direct measurement of the output power at the mouth of the horn is much less precise than the result obtained by the thermodynamic method, but within the limits of its accuracy, it provides a rough confirmation of the result obtained by the thermodynamic method.

Directivity Pattern

The third and last method of determining the efficiency involved a determination of the total output power through a measurement of the levels at a fixed distance of 45 feet from the siren, without the seven-foot extension horn.

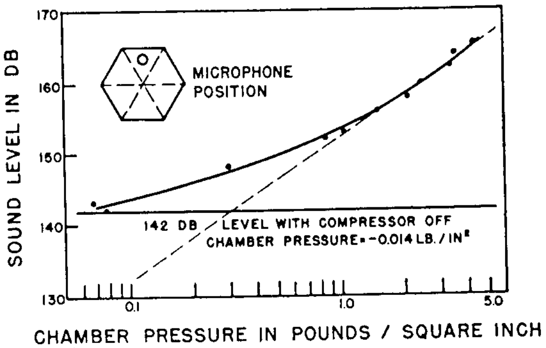

FIG. 6. Relation between chamber pressure and the sound level in the plane of the mouth of the extension horn. The dashed line shows the output level which would be obtained for lower chamber pressures if the sound pressure were directly proportional to chamber pressure.

Because of the lengthy measurements which were involved with this method, it was decided to measure the directivity pattern with the compressor inoperative. Figure 6 shows how the level depended on the chamber pressure. As the figure indicates, the pressure in the chamber is slightly negative when the compressor is not operating. The measurements indicated that the level dropped about 24 to 28 db when the compressor was stopped.

It was found, however, that the level was somewhat erratic when the compressor was off, and this effect was finally traced to the action of winds. The cure for this trouble was to close the intake of the compressor. When this was done, the negative pressure in the chamber increased to 0.10 lb./in.^{2}, and the output level was independent of the direction and strength of the wind.

The microphone was hung from a tripod 16 feet high so that the effect of ground reflections could be eliminated. The procedure was to vary the height of the microphone so that the heights and levels of the maxima and the minima could be determined. The minima were quite sharp, and the levels at the minima were from 10 to 15 db below the levels of the maxima. The positions of the maxima and minima agreed very well with those calculated by geometric methods using an amplitude reflection coefficient of 70 to 80 percent, and a phase inversion at the reflection from the ground. The ground was covered with thick grass in the area where the measurements were made.

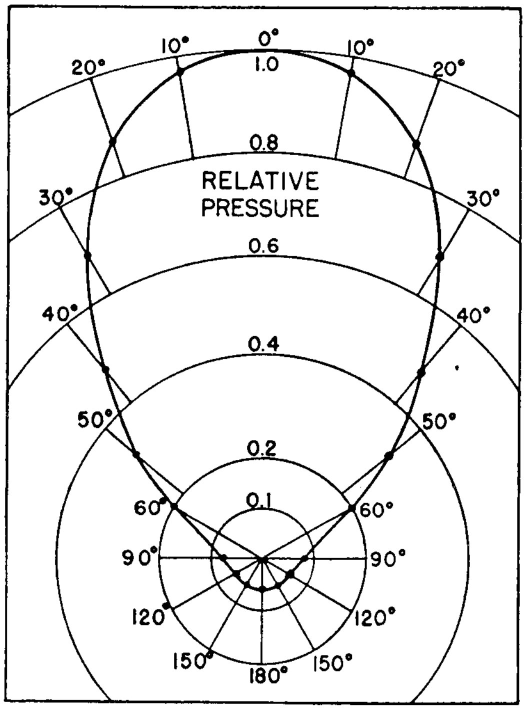

FIG. 7. The directivity pattern of the siren without the seven-foot wooden extension at 440 cycles per second. The pattern is corrected for ground reflections and thus represents the pattern which would be obtained under free field conditions. The polar curve represents the relative sound pressure as a function of azimuth at a constance distance from the siren. If the inverse square law be assumed, it also represents the shape of the contours of equal sound pressure, with the radius in the plot representing the distance from the siren. The directivity index of the pattern is 11.8.

The directivity pattern obtained in this way is shown in Fig. 7. The values there given have been corrected for ground reflections so that the pattern shows the directivity under free field conditions. It has a directivity index of 11.8.

Measurements were also made of the level with full output at distances of 5, 8, and 15 feet from the siren. These measurements indicated an effective source six inches behind the plane of the mouths of the exponential horns, and were extrapolated to indicate a free field level of 135 db at a distance of 100 feet.

A source of sound which produced a level of 135 db at a distance of 100 feet in every direction would represent a power of 369 kilowatts or 495 horsepower. With a directivity index of 11.8, the power represented is only 41.9 horsepower. This figure is to be compared with the 60 horsepower which was obtained by mapping the mouth of the extension horn.

The pattern method is believed to yield results that are less reliable than those obtained by mapping the horn, for the level fluctuations in the open sound field were found to be much greater, extending over a range of about five decibels.

The conclusion of the discussion of the third method is that the results are not inconsistent with those obtained by mapping of the horn opening, but they do suggest that the efficiency obtained by the latter method should be reduced.

Summary

The total output powers yielded by the three different kinds of measurement are 50, 75, and 52 horsepower, respectively. The result obtained

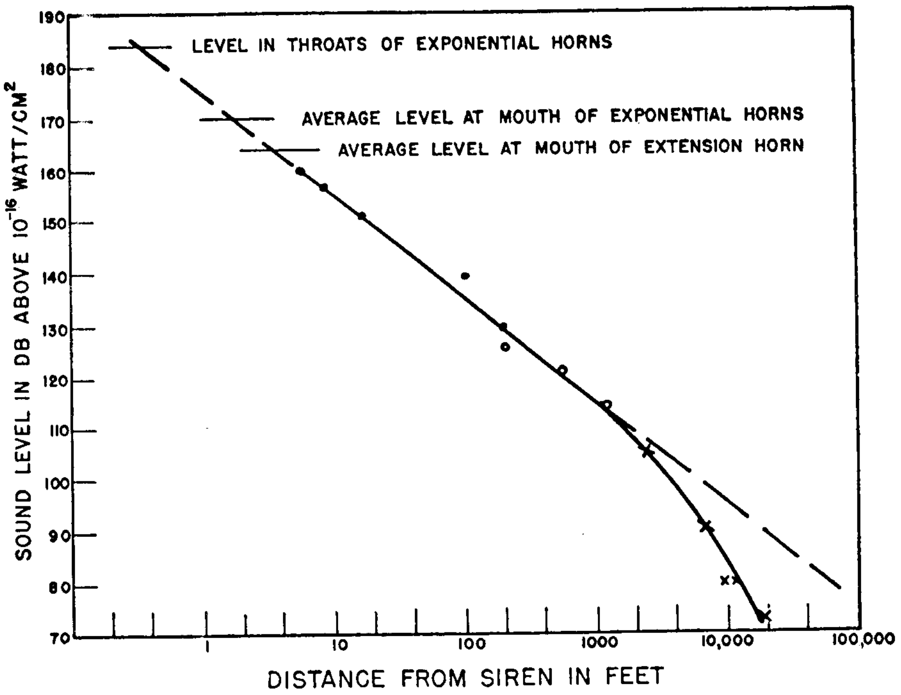

FIG. 8. A plot showing the maximum sound levels measured at distances from zero to 17,500 feet. The crosses represent measurements made with a sound meter in a boat in Barnegat Bay with a strong wind blowing against the direction of sound propagation. The open circles represent measurements in the marsh with a sound meter, and the solid circles show the levels measured with the shielded microphone. The levels in the mouths of the extension and the exponential horns, and also in the throat of the exponential horns, are also indicated on the plot.

by the second method is certainly too high. On the basis of a total input power of 69 horsepower (49 from the airstream and 20 from the rotor), the efficiencies are respectively 72, 109, and 75 percent. The first method is the most accurate, and the last the least accurate.

Dependence of Sound Level on Distance

During the tests, the sound level was measured at distances from the siren ranging from zero to more than three miles.

The path in the direction of the principal lobe of the siren's directivity pattern led over a marsh for about a third of a mile, and then led over the water of Barnegat Bay for a distance of six miles.

Figure 8 shows a synoptic picture of the levels obtained with the seven-foot extension horn. The levels shown are the *maximum* levels observed. At large distances, the level fluctuated over a range of about 20 decibels. On both of the occasions when measurements were made over the surface of the water, a 20 to 30 knot wind was blowing from the boat toward the siren.

The level of 170 db at the mouth of the exponential horn is calculated by energy considerations from the average level of 164 db at the mouth of the extension horn. Actual measurements of the level at the mouth of the exponential horns gave results at some points as high as 174 db.

The energy level of 184 db, at the throat of the exponential horns is calculated from a total sound power of 50 horsepower, or 37.3 kilowatts. Since the area of the horn throats is 22 in.^{2}, the power density must be 1695 watts/in.^{2}, or 263 watts/cm^2. The last figure corresponds to 184.2 db.

The Test in New York City

Because so little information was available about the propagation of sound in cities, particularly in areas with a number of tall buildings, it was decided to make a test of the siren as an air-raid warning device in New York City.

For this purpose, the truck with the siren mounted upon it was driven to the center of the Manhattan Bridge, and was placed in such a

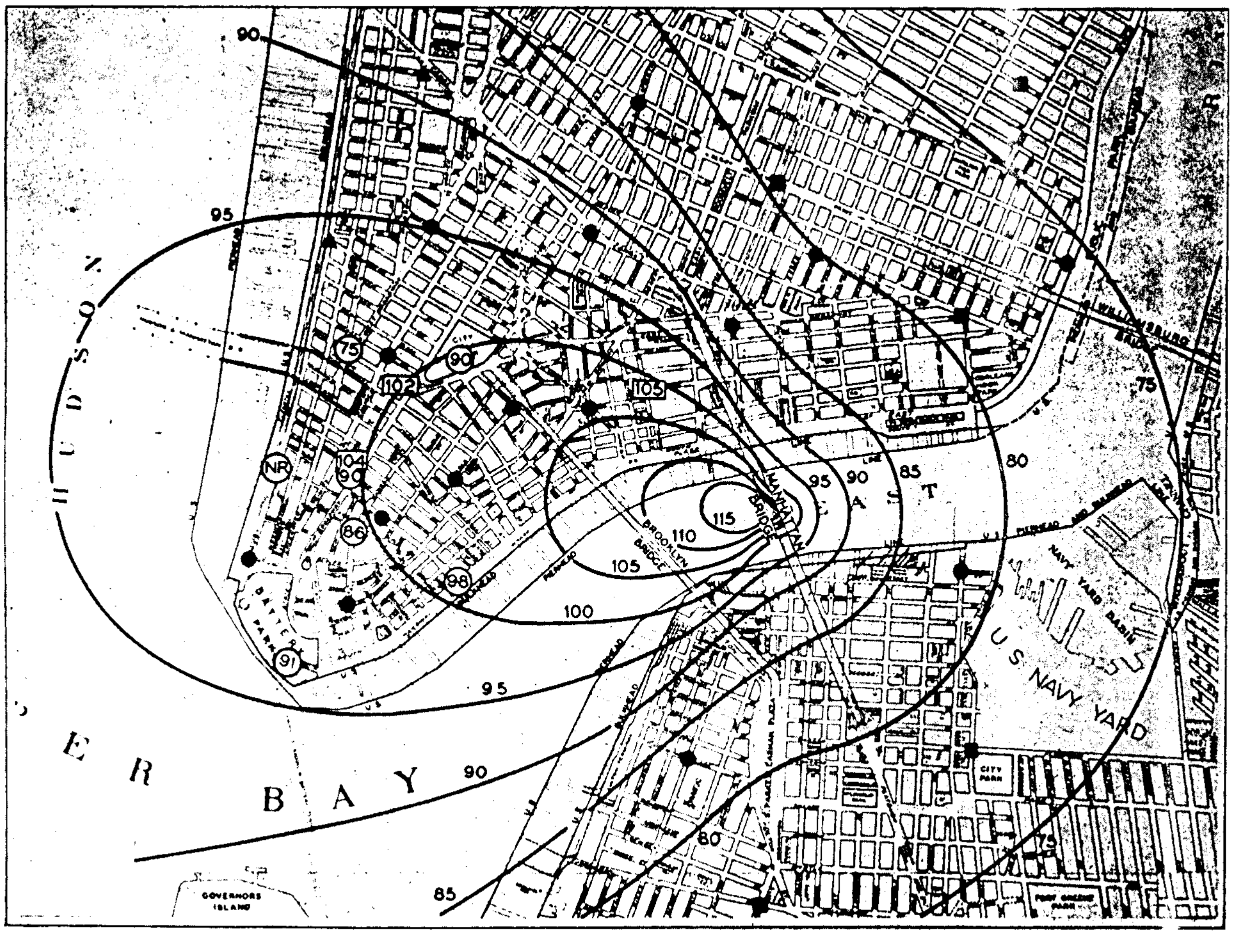

FIG. 9. A map showing the results of the test of the siren in New York City on March 4, 1942. The heavy lines show the contours of equal free-field sound pressure based on detailed tests in New Jersey, plus the assumption of the inverse square law. (Refer to Fig. 7.) The numbers within the open rectangles indicate the levels measured on the roofs of tall buildings, and the numbers within the open circles indicate the levels measured in the street. It will be noted that the roof-top measurements agree remarkably well with the expectation that they should be three decibels greater than the free-field contours, but that the levels in the street are very roughly fifteen decibels less than the contours, with the exception of the two observers who were not shielded by tall buildings. The map also shows the qualitative estimates made by members of the radio motor patrol service of the Metropolitan Police; the solid circles indicate an adequate signal, the solid squares indicate a faint signal, and the solid triangles indicate that the siren was not heard.

position that the major lobe of the directivity pattern extended in the direction of the financial district, as shown in Fig. 9.

Observers equipped with sound meters were stationed at the points indicated by the open circles and rectangles. The figures within the circles and rectangles indicate the sound levels which were measured. The rectangles refer to measurements which were made on the roofs of tall buildings, from which the siren was directly visible. The circles refer to measurements made in the street at ground level. It will be noted that, whereas the roof-top observers measured levels quite consistent with the contours which were drawn on the map on the basis of the inverse square law, the levels measured in the street appear to be roughly fifteen decibels less, with the exception of the two ground observers who were not shielded from the siren by tall buildings.

CONCLUSION

This paper has described the theory, design, and testing of a siren with a power output of approximately fifty horsepower and with an efficiency of approximately 70 percent.

The most important features of this siren from the point of view of realizing a high efficiency are an acoustically correct horn to provide the proper impedance for the ports to work into, a separate compressor to provide a large air flow at low excess pressure, ports having peripheral dimensions that are small compared with the wavelength, and a high ratio of blade width to part width.

The tests confirmed within the errors of measurement the results of the theory developed in Part I.

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A Powerful High Frequency Siren*

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This paper describes the design, construction and performance of a high intensity, high frequency (3–34 kc) siren and some rather striking phenomena which occur in the intense sound field produced by it. The siren is of the usual type consisting of a rotor which interrupts the flow of air through orifices in a stator. The rotor consists of a disk, approximately 6 in. in diameter, with 100 equally spaced slots and is driven by a small ⅔ hp high speed motor whose speed is varied by changing the applied voltage. The 100 corresponding holes in the stator are circular in section. For better radiation of the lower frequencies a plywood horn is mounted on the siren. The siren itself is small, and may be operated in any orientation. In its initial form, operating with chamber pressures in the neighborhood of 0.2 atmosphere, its measured efficiency was between 17 percent and 34 percent in the frequency range 3 to 19 kc with an acoustic output between 84 and 176 watts. There is a fair amount of parasitic noise, but it is negligible compared to the signal. With recent modifications, chamber pressures of about 2 atmospheres were obtained, yielding acoustic outputs of approximately 2 kilowatts, and an efficiency of about 20 percent.

INTRODUCTION

THERE has been a need in our work for a portable, directional, and trainable sound source capable of producing extremely high acoustic intensities and power outputs in air, over the frequency range of 3–30 kc. A survey of existing sources revealed none yielding sufficient power. A study of theoretical possibilities of various types led to the conclusion that the siren was most promising, especially in view of the success of the low frequency siren of the Bell Telephone Laboratories.[1]

The siren has been built and has proved to be a convenient, efficient and practical laboratory instrument. It can be operated in any orientation. During the past few months it has had about 200 hours of operation and is still in good condition. With ordinary precautions this appears to be but a small portion of its useful life.

Recently some structural changes were made in the siren to allow for higher air flow and pressure. A tenfold increase in acoustic output was obtained. However, since experimentation with this latest model has just begun, the discussion is restricted to the lower-power model, except in the last section which describes striking phenomena resulting from the high acoustic power produced by the latest model.

DESIGN AND CONSTRUCTION

Figure 1 shows the essentials of the siren in a cross section view. In any siren there are the following components: a source of air, a rotor which interrupts the air flow at the frequency of the sound desired, and ports in a stator through which the air escapes.

In the present case the source of air is a laboratory air line which connects to the chamber through three equally spaced air inlet tubes and supplies approximately 60 cu ft/min. The deflecting plate acts to distribute the air around the chamber.

The rotor is driven by a Dumore W2 motor which is rated at ⅔ hp with an input of 710 watts and a speed of 133 r.p.s. It has been operated at speeds as high as 340 r.p.s., under the load imposed by the siren, when the power input was approximately 1200 watts. Fortunately this higher speed provided increased cooling and no serious overheating resulted. The motor speed is easily controlled by a Variac when operated on a.c., or a rheostat when on d.c.

There are 100 conically shaped ports in the stator, equally spaced on a 6-in. circle. They are circular in cross section, 0.5 in. long, with diameters of 0.094 in. and 0.188 in., respectively, at the throat and mouth. The throat diameter is half the distance between centers. The total area of the openings is 4.43 sq. cm.

The rotor (Fig. 2) has 100 teeth, which are slightly wider than the ports to assure complete closing of all ports at the same time.[2] The ad-

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* This work was done under Contract No. W-36-039 sc-32001, U. S. Army, Signal Corps Engineering Laboratories, Bradley Beach, N. J.

[1] R. Clark Jones, J. Acous. Soc. Am. 18, 371 (1946).

[2] The same purpose would be met equally well if the port throat diameters were less than half the distance between

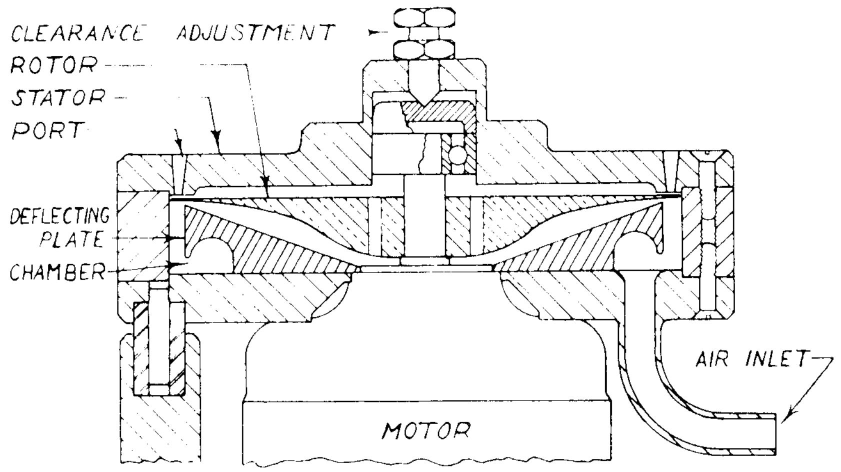

FIG. 1. Cross section of the siren.

jacent surfaces of the rotor and stator are ground plane and lapped so that extremely close operation (about 0.001 in.) is possible. This reduces wasteful air leakage when the ports are closed. Fine clearance adjustment can be obtained accurately by means of the screw, in the center of the stator, as illustrated by Fig. 1, which presses against a floating bearing on the end of the rotor shaft. This presses the rotor and shaft as a unit against a spring loaded bearing in the base of the motor, and provides positive, fixed, minimum clearance regardless of any expansion of the shaft caused by heating. Air passages through the rotor allow equalization of pressure on the two sides to prevent excessive loads on the bearings.

At high operating speeds stresses become important. Therefore, the rotor was made of the highest tensile strength aluminum alloy commercially available (Dural 75ST, tensile strength 80,000 lb/in.^{2}). Its shape follows the curve:

$$y = A \exp -bx^2,$$

where y is the thickness and x the radius. In a disk with symmetry about its mid-plane this shape gives uniform stress throughout the rotor.[3] The estimated safety factor exceeds the value of 10 at the highest speed of operation.

For frequencies of about 30 kc the ports themselves are good radiators. At lower frequencies better radiation is obtained by addition of the plywood horn shown in Fig. 3. It has an exponential taper, which for an infinite horn gives a theoretical cut-off frequency of 1 kc. Figure 4 is a photograph looking down into the horn. The siren ports are visible between the central and outer portions of the unit.

Figure 5 is a photograph of the siren without the horn. The height from base to top of adjusting screw is 11.5 in., the diameter of the chamber 7.5 in. It weighs 55 pounds. It has rubber mountings and is virtually vibration free.

Normally the siren is operated in a small anechoic enclosure. It is necessary to wear ear defenders of some sort when the door is opened during operation of the siren, especially at audible frequencies.

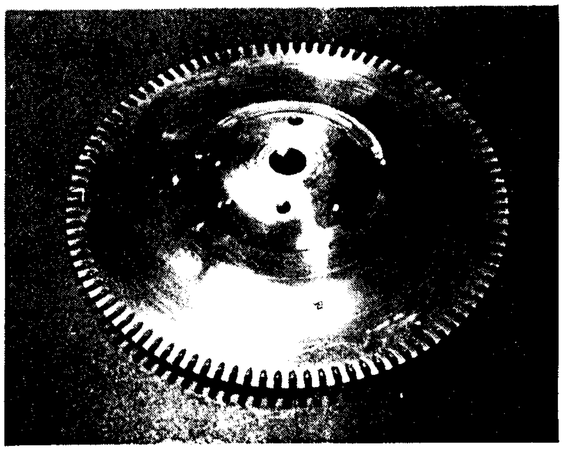

FIG. 2. Rotor.

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centers, and the rotor teeth were equal to the open space between the teeth. This would be preferable since it would result in greater efficiency (see reference 1). The present spacing is simply the result of modifications which had to be made on an earlier model.

[3] A. Morley, *Strength of Materials* (Longmans, Green and Company, New York, 1934), eighth edition, p. 388.

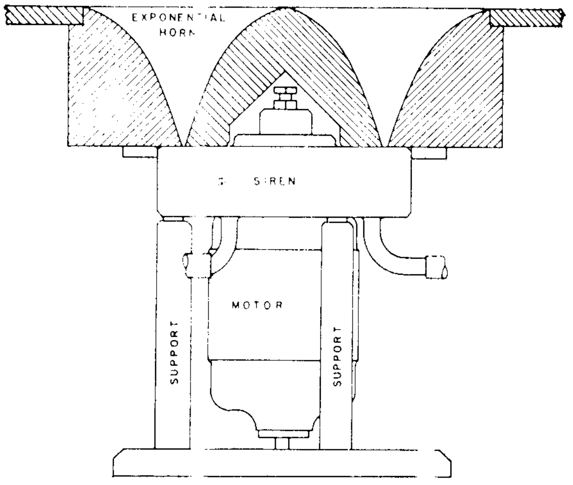

FIG. 3. Cross section of the plywood exponential horn in position on the siren.

PERFORMANCE CHARACTERISTICS

Acoustic Measurements

Sound intensities were measured with two different microphones. Where the field patterns were relatively broad, a Brush Standard Hydrophone (type BM-101) was used. Its rubber housing of the crystal is $\frac{5}{8}$ in. in diameter and $1\frac{1}{8}$ in. in length and it is calibrated to 60 kc for plane waves in air. On the other hand where the intensities were much less uniform or in narrow channels, as in the throat of the horn, a smaller Rochelle Salts crystal microphone, developed in this laboratory, was employed. Its dimensions are $\frac{3}{8}\times\frac{5}{16}\times\frac{1}{4}$ in., and it is calibrated in air to 80 kc. Below 30 kc it is essentially flat. Above that frequency its response drops and does not return to its lower frequency value.

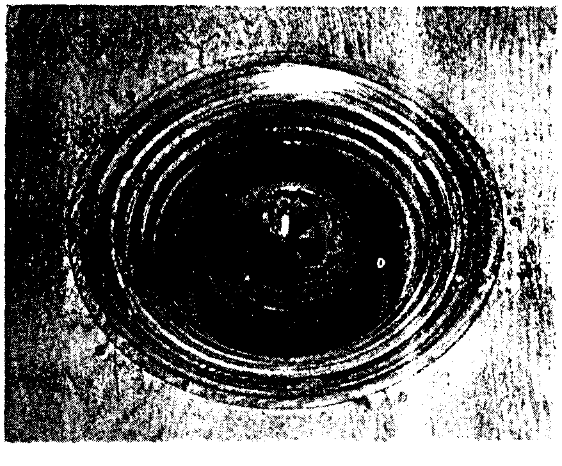

FIG. 4. Photograph of the horn in position. Siren ports may be seen in the throat of the horn.

Sound Patterns

The field pattern of the siren was investigated at several frequencies with and without the horn. The angular distribution of intensity (in db relative to 10^{-16} watts/cm^2) at a distance of 25 cm from the center of the siren face, and at a frequency of 3.25 kc, is depicted in Fig. 6. The peak intensity is 145 db; the points are experimental and the solid curve is the calculated directional pattern at a great distance from a ring source having a diameter of 6 in., the intensity being matched at the center. Most of the energy is in the central lobe. The heavy dashed lines show the geometric shadow cast by the adjusting screw.

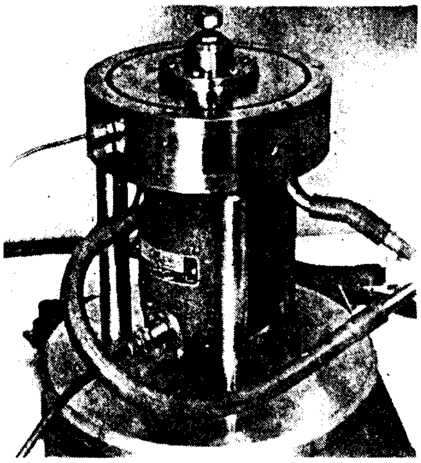

FIG. 5. Photograph of complete siren.

Figure 7 depicts the field pattern at the same frequency obtained with the horn in place. The solid line is a smooth curve through the experimental points. Here the peak intensity is seen to be increased 10 db and the central lobe broadened resulting in an increase in total output power greater than 10 to 1.

Results without the horn at 16.4 kc are presented in Fig. 8. The solid, calculated curve and the experimental points are seen to be in good agreement. Approximately 90 percent of the output power is contained in the first 40° on either side of the axis. At this frequency, the horn increases the axial intensity by only 3 db (see

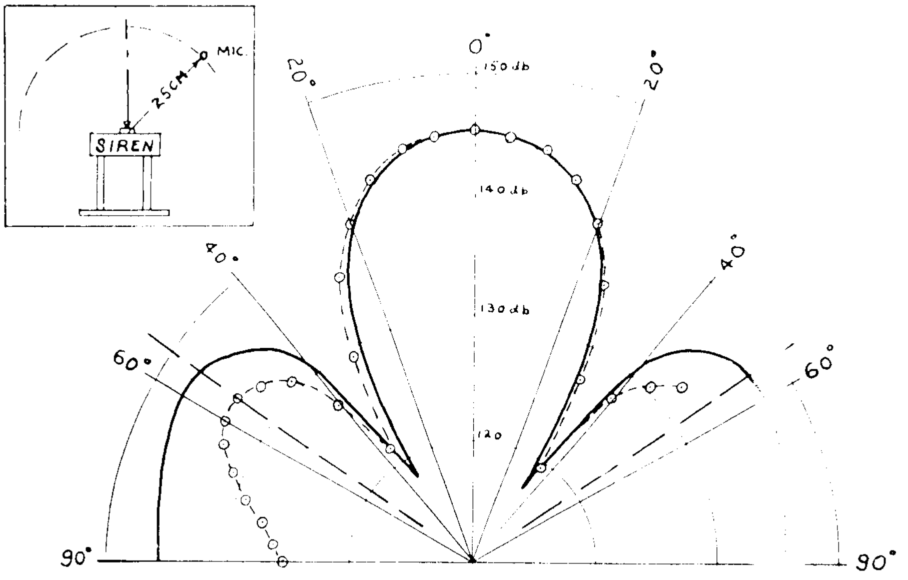

FIG. 6. Field pattern at 3.25 kc, microphone 25 cm from center of siren face. Solid curve, theoretical; dashed curve, experimental.

Fig. 9). The pattern is slightly broadened, but still closely resembles that for a ring source.

Efficiency and Power Output

The efficiency and power output of the siren with the plywood horn in place were measured by two methods: first, thermodynamically as described by Jones,[1] and second, acoustically, utilizing a calibrated microphone.

The data at a number of frequencies appear in Table I. They were taken at relatively low chamber pressures and volume flows, and hence the powe atputs given do not represent the maximum lues available.

nermodynamic Method

The tl iodynamic method, referrin to Jones' pa is based on the assumption hat the only f of energy, other than heat, nto which the ailable energy of the air strea is converted acoustical energy. If all the a ailable energy ere used in this fashion, the m-perature of the exhausted air would be less an the temperature of the air in the chamber b the

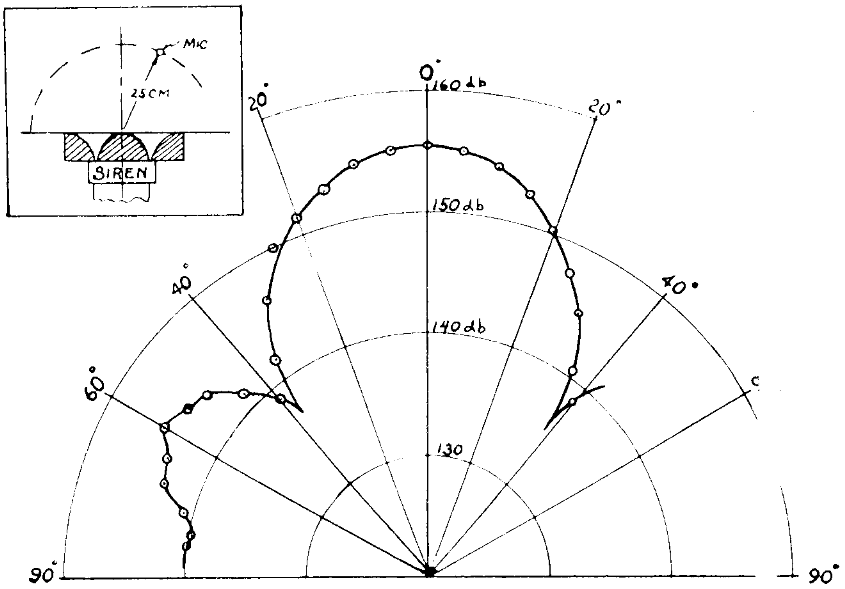

FIG. 7. Experimenta ield pattern, with horn, 3. kc, microphone 25 cm from ter of horn.

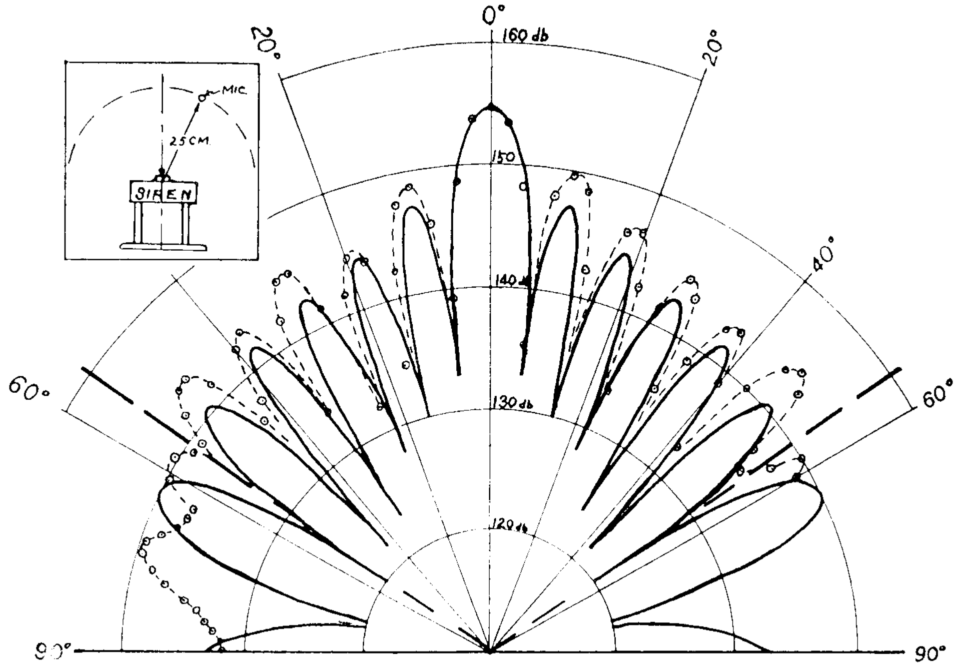

FIG. 8. Field pattern, 16.4 kc, microphone 25 cm from center of siren face. Solid curve; theoretical; dashed curve, experimental.

amount corresponding to an adiabatic, reversible expansion from the chamber pressure to atmospheric pressure. Thus the efficiency of the siren is the ratio of the measured temperature difference to that corresponding to an adiabatic reversible expansion between the two pressures.

The basic quantity to be measured then, is the difference between the temperature of the air immediately below and above the rotor, as illustrated in Fig. 10. The vertical separation between the internal and external thermocouple was kept small, about $\frac{1}{16}$ in., in order to minimize the effect of the dissipation of rotor energy in the form of heat. This was relatively important since at the high frequencies the rotor power requirements were large. Thus at 3.3 kc, 60 watts

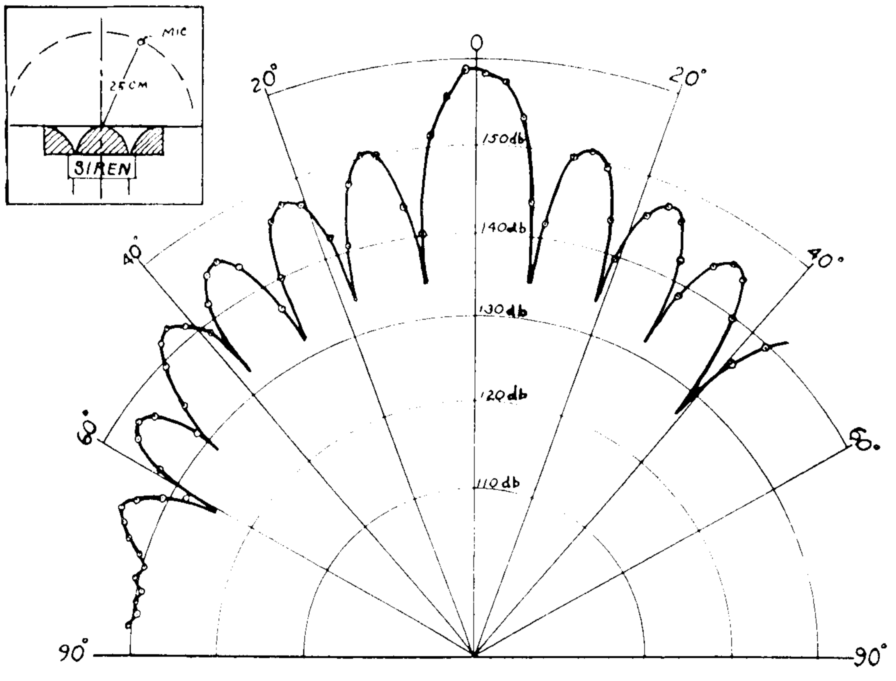

FIG. 9. Field pattern with horn, 16.4 kc, 25 cm from center of horn.

TABLE I. Acoustic Output and Efficiency Measured Acoustically and Thermodynamically at Various Frequencies.

Frequency (kc)	Chamber pressure (atmospheres)	Volume flow (cc/sec ×10³)	Power Input (watts)	Power output from acoustic measurements (watts)	Percent efficiency acoustically	Percent efficiency thermodynamically
3.25	0.18	2.33	433	125	29	32
5.75	0.20	2.47	500	146	29	31
6.75	0.22	2.46	516	166	32	29
7.3	0.22	2.45	513	176	34	32
8.8	0.22	2.47	507	131	26	29
10.7	0.19	2.46	446	103	23	27
11.5	0.18	2.46	441	95	21	25
14.6	0.16	2.44	392	101	26	20
18.9	0.20	2.48	488	84	17	25

of electrical power was used in running the motor and rotor; at 10 kc, 160 watts; at 18.9 kc, 350 watts; and at 34 kc, 1200 watts.[4]

Despite the precaution of keeping the thermocouples close to each other, the heating produced by the rotor reduces the temperature difference, especially at the higher frequencies. In fact at 33 kc this source of error is sufficiently large to lead to a reversal in the sign of the temperature difference.

One further source of difficulty is the fact that the temperature indicated by the junction in the ports is a function of its position. The temperature measured is lowest in the axial region and increases considerably as the junction approaches the walls. This is due in part to the fact that the stator is warmer than the air entering the ports and consequently heats it and also radiates heat to the thermocouple. In all cases the temperature was measured in the axial region.

Acoustic Method

The acoustic determination was carried through by measuring the level in the horn at points 6 cm above the stator and integrating over the area at this height. The small microphone was used.

Observations with the microphone (by oscillograph and harmonic analysis) emphasized an important feature of high intensity, high frequency sources and waves. For very high intensities, corresponding, for instance, to the powers listed in Table I, a wave is subject, while being propagated, to finite amplitude effects which cause it to change shape, one form toward which it tends being "saw tooth."[5] The distance of propagation in which this is effected decreases with increasing intensity and frequency. We have found that above 8 kc this change is practically complete in the 6 cm of propagation in the horn. Thus the nature of the propagation of high amplitude waves is such that for high frequencies, regardless of the purity of the initially produced wave, at relatively short distances from the source the wave will be rich in harmonics. On the other hand for low frequencies, the waves may be very nearly pure for large distances.

In obtaining the data of Table I, the levels were read from a flat amplifier without correction for the frequency characteristic of the microphone. In view of the high harmonic content of the waves due to the finite amplitude effect it is clear that these readings yield lower intensities than actually exist in the s d wave. This factor operates independently of and in addition to any attenuation caused by increased absorption of the higher harmonics which are produced when the wave shape becomes saw-toothed. It is our

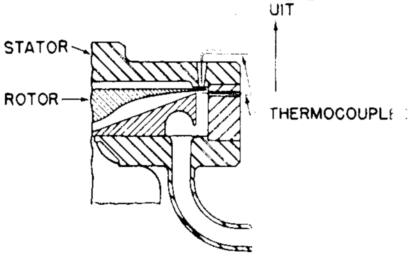

FIG. 10. Section view showing the position of the thermocouples for thermodynamic measurements. The vertical separation is about $\frac{1}{16}$ in.

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[4] It should be pointed out that the thermodynamic method gives the efficiency of conversion of the energy in the air stream into sound energy and does not take into account the energy used in running the rotor. This measure of efficiency can be justified on two accounts. In the first place, as the air stream power is increased the rotor power becomes a smaller part of the total (for example at 18.9 kc with an air stream power of 488 watts the power to run the motor and rotor is about 72 percent of this value, to be compared with 8 percent when the air stream power is 9.5 kilowatts). Thus in the region in which the output is a linear function of the input, the overall efficiency (including the energy going into the motor) approaches that calculated. Secondly, the power required to run the motor bears much the same relation to the siren operation as field coil power does to the operation of an electrodynamic speaker: the power sources are both cheap and highly accessible and are therefore of little consequence.

[5] R. D. Fay, J. Acous. Soc. Am. , 222–241 (1931).

elief that this, more than any other single item, mits the production of a pure signal of high ntensity and frequency at a reasonable distance rom a sound source. It is regrettable that there s a paucity of theoretical and experimental work n this subject.

It would seem that for high frequencies, both he acoustic and thermodynamic procedures lead o lower calculated efficiencies than probably ctually exist. This may well explain the drop in fficiencies in Table I as the frequency is inreased. In view of all these considerations, the greement between the two methods is better han could be hoped for.

BSERVATIONS IN THE INTENSE SOUND FIELD

The following results were obtained with the iren operating at the levels given in Table I. he frequency most used was 7000 c.p.s.

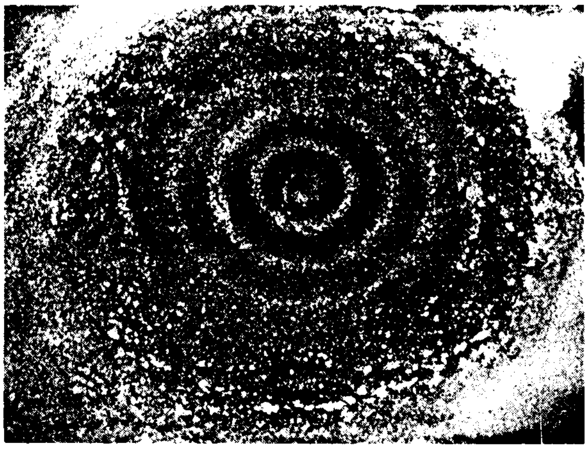
Sonic Wind and Pressure

As was seen, the radiation pattern of the siren, ith or without the horn, has a pronounced entral lobe. It has been found that in this entral region there is a relatively strong current f d.c. air flow away from the siren. This "sonic vind" is associated with the acoustic wave ather than the d.c. flow of air from the siren. To demonstrate this the latter was deflected out of the sound field by means of screens made of heese cloth. With these in place, the wind velocity in the central region was measured with full air flow, both when the siren was rotating, and when it was stationary with the ports fully open. In the latter case air velocities were well below 1 m sec, while with the siren in operation velocities as high as 7 m sec. were measured. Moreover the sonic wind was greatest in the regions where the acoustic intensity was maximum. It is obvious that since the sonic wind is not associated with the d.c. supply of air through the siren, the air must come from points outside the siren and outside the axial region. Indeed this was confirmed by direct observation. The flow pattern thus resembles a spreading fountain, the air flowing up the axis, then curving outward and descending at some distance from the axis again to be fed into the central spout, the energy to keep the fountain going being supplied by the sound.

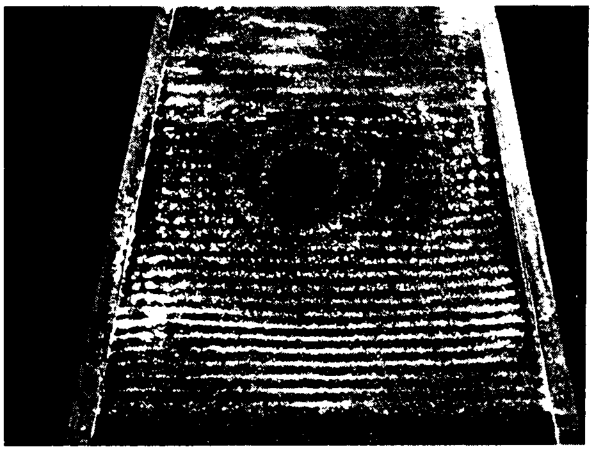
Associated with the sonic wind, there were significant departures, both positive and negative, of the mean pressure from atmospheric. The data are too meagre for any conclusions to be drawn, suffice it to say that deviations of more than 100 dynes cm^2 have been measured.

Dust Patterns

Properties of the direct radiation pattern, such as are shown in Figs. 6 9, can be rendered visible by means of dust particles. In one such experiment fine sawdust was sprinkled on a cloth screen supported parallel to the face of the siren and approximately 15 cm above the face of the horn. The sawdust then collected in concentric rings, demonstrating clearly the circular sym-



(a)



(b)

FIG. 11. Dust patterns on horizontal silk screen 15 cm above the horn, about 20 kc; (a) in free field, (b) superposed interference pattern when a plywood reflector is inclined to the screen away from the observer.

metry of the radiation pattern. Figure 11(a) is a photograph of the pattern obtained with a 22 kc signal. The interference pattern produced by the direct radiation and that reflected from a plywood reflector, whose plane is at an angle of about 35° relative to the siren face, the screen being in the same position as in the previous figure, is shown in Fig. 11(b).

The formation of patterns in this manner was possible only after the screen was given a thin coating of graphite to make it electrically conducting. Prior to this, such high electrification occurred almost immediately, that the sawdust would cling to the screen and not move. After the sound was shut off, the charge persisted so that the screen could be inverted and shaken without dislodging the sawdust.

Burning Sensation

Another effect which was noticed early was the burning sensation produced when the outstretched hand is held, with the fingers together, in the radiation field. The heating is felt to occur at those parts of the fingers which are close but not quite touching. This occurs almost immediately, but may be relieved by spreading the fingers. Otherwise a painful burn may result.

This burning effect is associated with a real temperature increase as was indicated by taping a thermocouple to a finger. The effect could be simulated by using soft rubber tubing in place of the fingers. Temperature increases of as much as 45°C were measured. On the other hand the effect could not be duplicated with stiff rubber tubing or with glass rods—the temperature rise in these cases was less than 1°C.

Finally, in an ambient sound field of 165 db, the crystal microphone inserted between two fingers read 178 db when the burning sensation occurred.

While the data are inconclusive, we are inclined toward the working hypothesis that the heating is produced by the damping of vibrations, not necessarily of the same frequency as of the sound, which are set up in the fingers by the intense sound field in the region between them.

Phenomena in Standing Waves

A standing wave of sorts, can be obtained by setting up a suitable reflector in the sound beam. This is accompanied, of course, by an increase in sound pressure level. Levels of about 170 db have been measured. Some rather striking effects occur similar to those observed by St. Clair[6] using an electromagnetic sound generator. Coins, small balls, and other small objects float at the velocity antinode surfaces, being supported only by the acoustic radiation. It is interesting that objects floating at the same or neighboring level tend to attract each other. Two balls in the same antinode surface float in constant contact and when they are in neighboring levels the upper ball will follow the lower on its wanderings in the field, each staying at its respective velocity antinode surface.

Heating by Acoustic Absorption

As has been mentioned, intensities of 160 db are available in the free radiation field. This corresponds to 1 watt/cm^2, an energy flow so large that it should cause a measurable rise in temperature in an absorbing material. With this purpose in mind a half pint thermos bottle was filled with absorbent cotton. When its open end was exposed to a 7 kc signal the cotton caught fire in 17 min. Incidentally, after smothering the fire the cotton was found to have been burnt in the first inch only, indicating dramatically that most of the acoustic energy was absorbed in traveling through 1 in. of cotton.

The attempt was then made, actually to measure the intensity of such sound fields in terms of the rate of temperature rise in an absorbing medium of known heat capacity. A thermos bottle was therefore filled with copper turnings packed to form a good absorber. The resulting system was very close to thermal equilibrium at all times. With a temperature rise of 3.5°/min., a capacity of 22.6 cal/deg, the power entering the bottle was calculated to be 5.6 watts. Measurements with a microphone gave 5.9 watts. Subsequent tests indicate that this experiment was conducted at nearly the optimum intensity for best results. At much lower intensities radiation and convection losses become significant while at much higher intensities new phenomena are introduced which require a more elaborate technique than the simple one outlined ab : in orde to obtain reliable results. Since th new phenomena are not yet entirely underst a discussion of this type of measurement at higher intensities will be delayed until a lat paper.

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[6] H. W. St. Clair, Rev. Sci. Instr. 12, 250 (1941).

E ERIMEN S WITH HIGHER POWER SIREN

e preceding discussion has concerned itself en ely with the siren as originally constructed, op ating at a chamber pressure of about 0.2 at sphere and air flow of 2.5×10^4 cc/sec. This sec on is concerned only with the latest siren for wh h an acoustic power output exceeding 2 kw, wi an efficiency of about 20 percent, has been m sured. We propose to study this higher power si in considerable detail and to report on it la . At present we shall give only a brief desc ption of some of the rather striking phenc nena observed with it.

he suspension of objects in a standing wave fie may be demonstrated rather spectacularly w the increased power. Thus it has been possi to float simultaneously as many as seven gl s marbles, ¾ in. in diameter, n a single a inode surface. The behavior of the spheres is in resting. They cling together and spin as a g up with increasing angular velocity, about t central one in a horizontal plane, until they fl apart. If this spinning motion is inhibited, for e mple by touching the group with a stiff wire p be, they stay in the field indefinitely. It is si nificant that it is also possible to float such s eres in the field in the absence of he reflector — never achieved with the lower power unit. The condition is unstable, however, unless provision is made to keep the spheres centered.

Similar interesting results occur when coins a d other objects are "floated." No systematic in estigation of their behavior has been attempted—most of our observations aving been n de simply for amusement. When number of pennies are placed on a stretched sil screen, the parameters can be so adjusted that the pennies do somersaults with "Rockette"-like precision; or so that a penny can be made to se slowly to a vertical position, appearing all he while to be supporting, acrobatically, another which finally assumes a horizontal positi above the first penny and touching rim to rim. Also coins resting on the silk screen can be flipped a distance of a few feet by varying the frequency of the siren rapidly.

As further demonstration of the high acoustic power, it has been possible to burn a cotton wad in 6 sec. or steel wool in about a minute by simply holding it in the hand a short distance above the siren. Moreover, a sample of Quietone placed behind 1 in. of very low density Fibreglas caught fire in 45 sec. The Fibreglas was necessary in order to reduce air circulation. To satisfy a skeptical colleague his pipeful of tobacco was lit without benefit of match or puffing by simply exposing the open end of the bowl to the intense sound for a short time.

Finally, it may be recalled that M. Y. Colby[7] in his textbook on acoustics asks how long one would have to talk to a cold cup of coffee in order to heat it for drinking if all the sound energy striking it were converted into heat. The answer is between 75 and 20,000 years depending on how close one gets to the coffee. By way of a stunt it was decided to go a step further and find out how long the siren would have to talk to a coffee pot in order to brew a cup of coffee. In 7 min. the water boiled up in the makeshift "Silex" and a hot cup of black coffee was obtained. With a more efficient "stove"—an acoustic stove consists merely of material which absorbs sound—the coffee could have been brewed in a shorter time.

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It is with pleasure that we acknowledge that to the guidance and inspiration of Dr. H. K. Schilling, director of the Acoustics Laboratory in which this work was done, must be attributed a large part of what success may be claimed.

We should like also to express our thanks to Mr. L. J. Sivian of the Bell Telephone Laboratories, Dr. R. C. Jones of the Polaroid Corporation, and Dr. E. G. Pickels of the Rockefeller Foundation for helpful discussions on the design of the siren.

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[7] M. Y. Colby, *Sound Waves and Acoustics* (Henry Holt and Company, New York, 1938), p. 206.

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November 7, 1992
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ACOUSTIC RADIATION TEST USING AND ELECTRIC GUITAR

Acoustical radiation forces have been shown to be able to levitate objects. The utilization of this force for the purpose of levitation is primarily dependent on the lower frequency sonics in a longitude mode; they provide a lifting force in the form of pressure wave harmonics.

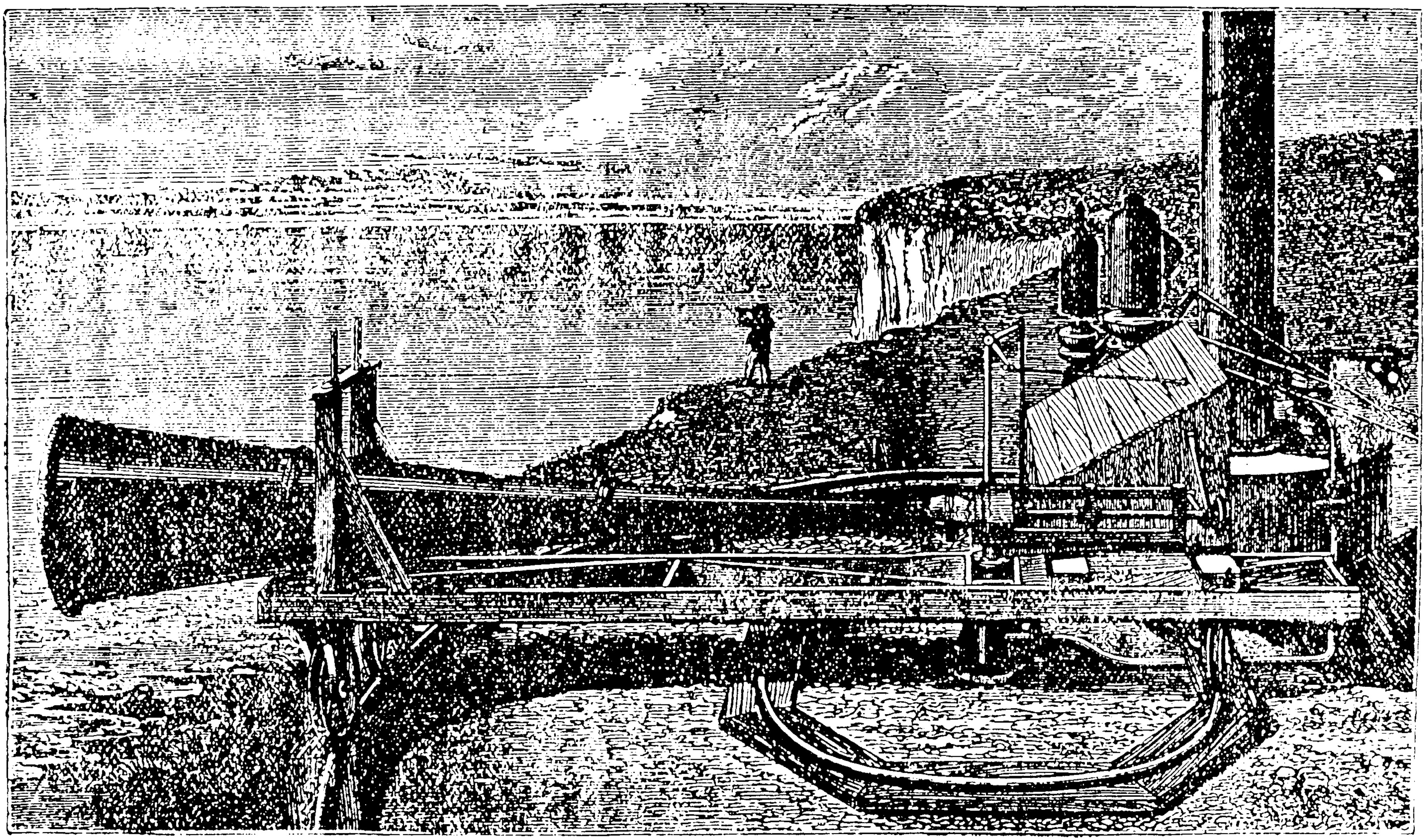
As shown in Figure 1, when the first fret E-chord on a guitar was struck through a distortion amplifier a state of motion was induced and minimal levitation sustained for up to 5 seconds at approximately 25 watts.

Higher frequency chords gave only vibrational responses. The distorted oscillation of the guitar strings can induce propulsion due to the pulse of many sonic waves set in motion.

In one test, shown in Figure 2, we proved that the harmonic frequencies of a guitar can indeed lift a speaker as high as a centimeter above the ground.

We have also discovered that the lower the frequency notes, less amplification was required to sustain levitation. This suggests that levels below hearing in the 30 Hz range may produce the greatest results at lower amplification.

On a flat surface, motion usually goes in the direction of highest weight offset. The weight tilts the speaker to change the direction of pressure force. In order to gain sufficient lift, high amplification is required. I am unable to conduct adequate experimentation due to the overheating of speaker coils! No complete analysis of the situation has been preformed.



Fog-Siren